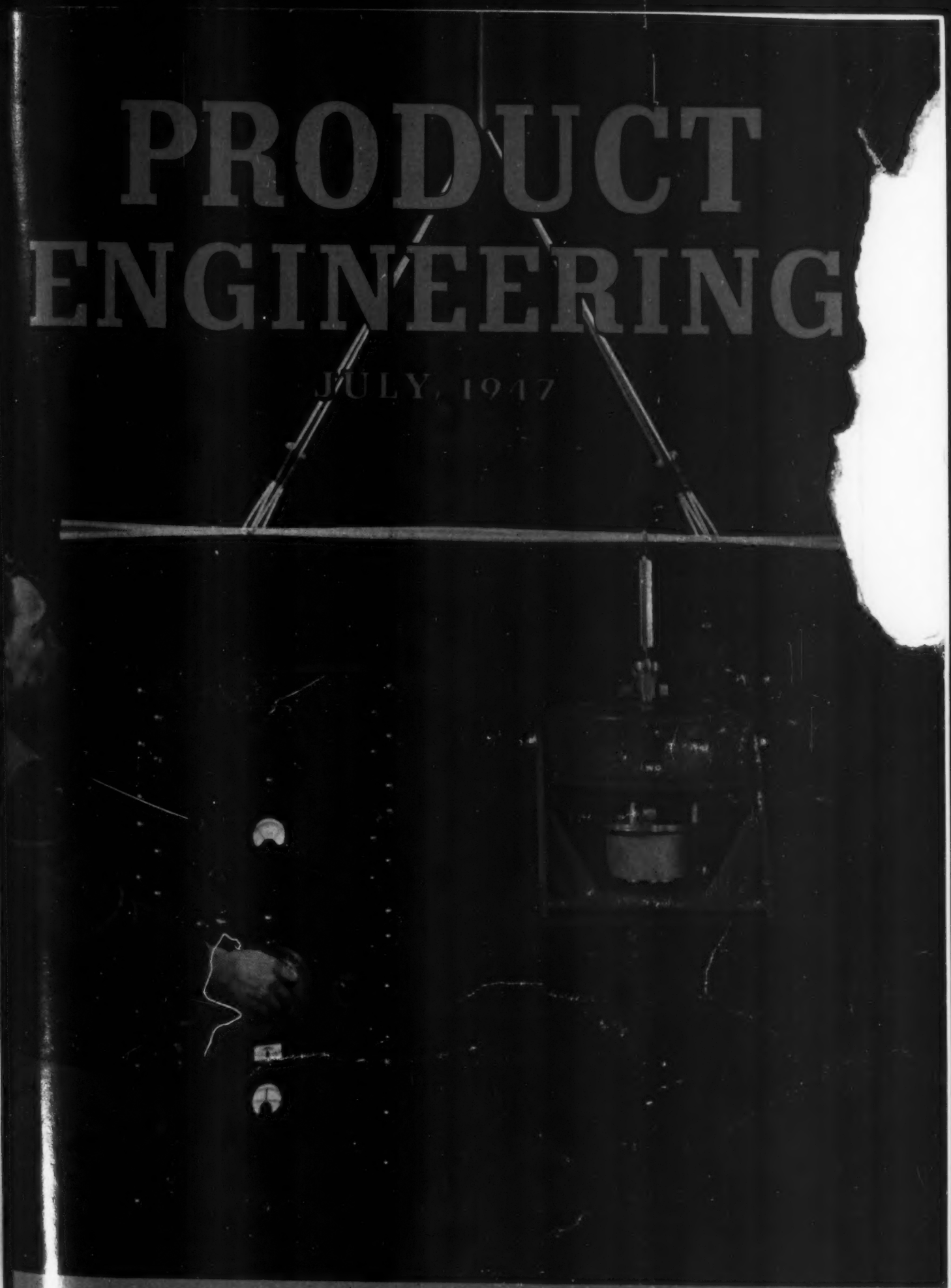
PRODUCT ENGINEERING

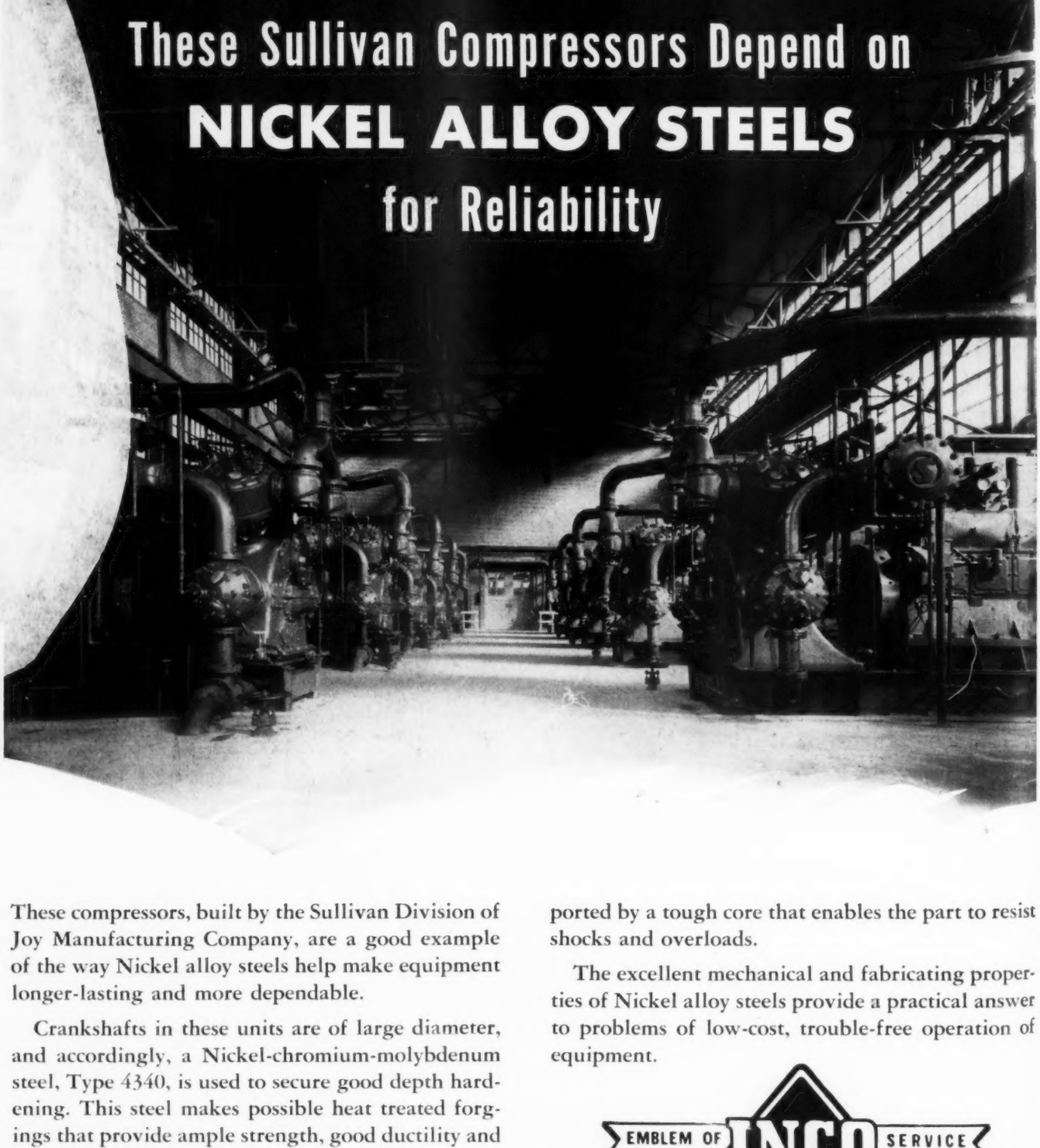
JULY, 1947



IN THIS ISSUE — Vibration Testing + Exploring New Markets + Constant Velocity Universal Joints + Designing for Zero Backlash + Fluid Flow Friction Factors

A McGRAW-HILL PUBLICATION

These Sullivan Compressors Depend on NICKEL ALLOY STEELS for Reliability



These compressors, built by the Sullivan Division of Joy Manufacturing Company, are a good example of the way Nickel alloy steels help make equipment longer-lasting and more dependable.

Crankshafts in these units are of large diameter, and accordingly, a Nickel-chromium-molybdenum steel, Type 4340, is used to secure good depth hardening. This steel makes possible heat treated forgings that provide ample strength, good ductility and ready machinability.

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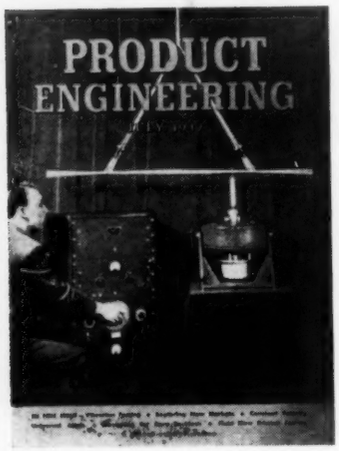

FFRONT COVER

TESTING the effects of forced vibrations on the behavior of a steel beam. McGraw-Hill Kodachrome, courtesy of Vibration Division, MB Company.

Vibration Testing to Improve Design

Many engineers are aware of the need for hard-to-get data concerning what really occurs in the machines they design. Even after operating defects are perceptible in the best designed machines, often the engineer can only guess what the contributing causes might be. On page 115, John A. Dickie, design engineer of the MB Manufacturing Company emphasizes the need for such data and tells how much of it can be obtained by shake-testing the machines and structures with vibration exciters.

New Products for Expanded Plants

Many companies repeatedly face the problem of finding new products to manufacture profitably. Beginning on page 85 is the story of how Warner & Swasey solved this problem by organizing a committee of engineers and salesmen, and, assigning the problem to them as a full time job.

Zero-Backlash Devices for Gear Drives

Excessive backlash in gear trains is the cause of erratic operation in many machines. Kenneth E. Lofgren of Cooper Union School of Engineering, describes on page 150 the effect of backlash in several types of gear drives and presents some simple methods for taking up backlash.

Electrical Analogies

Many engineers have discovered that an apt analogy does much to clarify a scientific principle in the minds of interested or important people who are not technically trained. On page 126, Record Rogers of General Electric Company describes some pet analogies that he has found useful from time to time.

Fluid Flow Friction Factors

Much more experimental work is needed to aid engineers in making estimates of the loss in fluid flow resulting from pipe friction, and obstructions of valves and fittings. The only practical source from which to obtain such information is basic research. V. L. Streeter of Armour Research Foundation of Illinois Institute of Technology, on page 89, discusses methods of determining losses in fluids through pipes and fittings, and methods for expressing these losses in a useful form.

Luminous Materials

During the war every field of science and engineering was searched for aids to night operation. Among the things found to give useful illumination without a power source were phosphorescent, fluorescent and self luminous radium-activated materials. On page 144, Dr. L. H. Dawson, Optics Division, Naval Research Laboratory, Washington, D. C., discusses properties and applications of these materials.

Constant-Velocity Universal Joints

Millions of constant-velocity universal joints were used during World War II in Jeeps and heavy trucks. They are being used today not only in the postwar Jeeps, but also in various types of automotive vehicles and industrial machinery. An article by E. B. Sturges, chief engineer, Universal Joint Section, Bendix Products Division, beginning on page 120, presents information to guide designers in applying universal joints to other products.

Basic Standard Drafting Practice

Interpretation of production drawings is one of the frequently discussed and at the same time unsettled problems in the field of engineering. On page 137, George E. Rowbotham, project engineer, Fisher Body-Ternstedt Div., General Motors Corporation, and also chairman of the General Motors Drafting Manual Subcommittee, presents a plan for compiling and adopting a nation-wide basic standard drafting practice that could be commonly accepted by many industries.

Hydraulic O-Ring Seal Packings

Under the impetus of demands made by military aircraft hydraulic systems, the O-ring packing was developed during the war, to save weight and fluid. This packing is now being used in many industrial applications. On page 163, Robert E. Allen, Plastic and Rubber Products, Inc., describes groove designs for stationary and moving types of O-ring seals in hydraulic units, and several arrangements for sealing cylinders, pistons, end caps and adjustable threaded parts.

Multimotor Drives

Adjustable-voltage control for multimotor drives is often used where a group of motors must operate together and stay in a desired speed relationship over a rather wide speed range. The advantages of controlling groups of motors that must operate together by some form of adjustable-voltage system are explained and discussed on page 167, by J. E. Jones, Supervising Engineer of Cutler Hammer, Inc. Basic circuits, different arrangements of motors, range of speeds, methods of keeping speeds together and some limitations are included.

AUGUST HIGHLIGHTS

Effect of High Pressure on Properties of Materials, by T. C. Poulter, Armour Research Foundation. Changes produced in physical properties of materials by a decrease in temperature and an increase in pressure are correlated.

Urea and Melamine Resin Products, by M. H. Bigelow and J. A. Murray, Plaskon Div., Libbey-Owens-Ford Glass Company. Properties, applications and design factors of molded melamine and urea plastics are discussed, as well as other resin applications, such as laminates and adhesives. Also production techniques, as related to product design.

Simple 8 to 1 Speed Range Motors, by R. W. Moore, Westinghouse Electric Corporation. Two separate magnetic circuits in the field do the trick. How the system works, performance data, and application factors are included.

PRODUCT ENGINEERING Editorial

JULY, 1947

Intangible Costs Often Outweigh Direct Costs

A DECISION against the use of a new material or process is often based on the relative direct labor and material costs between the old and the new. Although the judgment leading to this decision is often correct, in spite of the fact that it is based on incomplete data, labor and material plus overhead are never the only items that make up the total cost of a part or product. Numerous intangible items, some of which can not be evaluated directly in terms of dollars and cents, enter into every cost analysis.

Unfortunately many manufacturing cost items are lumped together under the term "overhead", which in turn is established as a percentage of direct labor and material costs based on past experience. This is a simple and convenient method of cost accounting but may be highly misleading when used to compare the true costs of proposed and existing designs. The dollar cost of purchasing, stores keeping, inventory and similar items will often be misleading figures if taken on the basis of percentages of labor and material cost.

Castings made of aluminum, as compared to iron or steel, cost considerably less to transport and to handle in the shop. A redesign in which several parts are united in a one-piece die-casting shows an appreciable saving in materials ordering, stores costs and perhaps in inventory. A standardized design that replaces several slightly different parts may cost more per unit of labor and material but be more economical because of larger lot production, lesser inventories, less cost of stores handling, paper work and purchasing. Many savings in cost of storage, handling and maintenance of patterns can be had by replacing castings with weldments. A small variety of steel shapes and plates will cover a large number of weldment designs; whereas it would require several of every size and shape to stock castings.

The cost of obsolescence is one that is often overlooked when analyzing the economics of a design. By standardization, minimizing the number of parts and reducing the patterns, dies, jigs and fixtures to a minimum, obsolescence costs will be diminished.

The most intangible cost of all, loss of buyer acceptance, must be carefully considered and weighted. Without question, this is the most important of the intangible costs. Pride of ownership will impel the prospective buyer to scorn the product that does not, in appearance and performance, bespeak quality, sturdiness, attractiveness and progressiveness.

Relative costs of two or more designs cannot be accurately determined unless every tangible and intangible cost that enters into the problem has been studied and weighed. The final decision as to which design is to be adopted must be based on a composite of the total costs and the relative sales values.

G. F. Jordenholt

Simplify and Save on Hydraulic Systems with VICKERS POWER UNITS

The "hydraulics" on this Ingersoll 24-Station Process Machine transfers the transmission castings from station to station and provides power for clamping, locating and chip disposal. The hydraulic system design was simplified and cost reduced by using the twelve standard Vickers Self-Contained Hydraulic Power Units shown.

These units provide a compact hydraulic power package which can be applied to almost any system. They are available with various pump combinations to meet a wide range of requirements up to pressure demands of 2,000 psi. There are 92 standard sizes and types of single stage, two stage and dual pressure pumps equipped with motors from ¾ to 20 hp.

The oil reservoir serves as a base to save floor space. On it is mounted a standard Vickers Motorpump or a Vickers pump with separate motor drive. A Vickers "balanced piston type" pressure relief valve provides accurate control of hydraulic pressure. A simple wrench adjustment sets maximum operating pressure at any desired figure within recommended pressure range. Filters and reservoir are readily accessible for cleaning. See Bulletin 46-43 for the many other features.

Vickers Hydraulic Power Units are frequently the most desirable source of hydraulic power. Vickers application engineers will be glad to discuss your hydraulic power and control problems.

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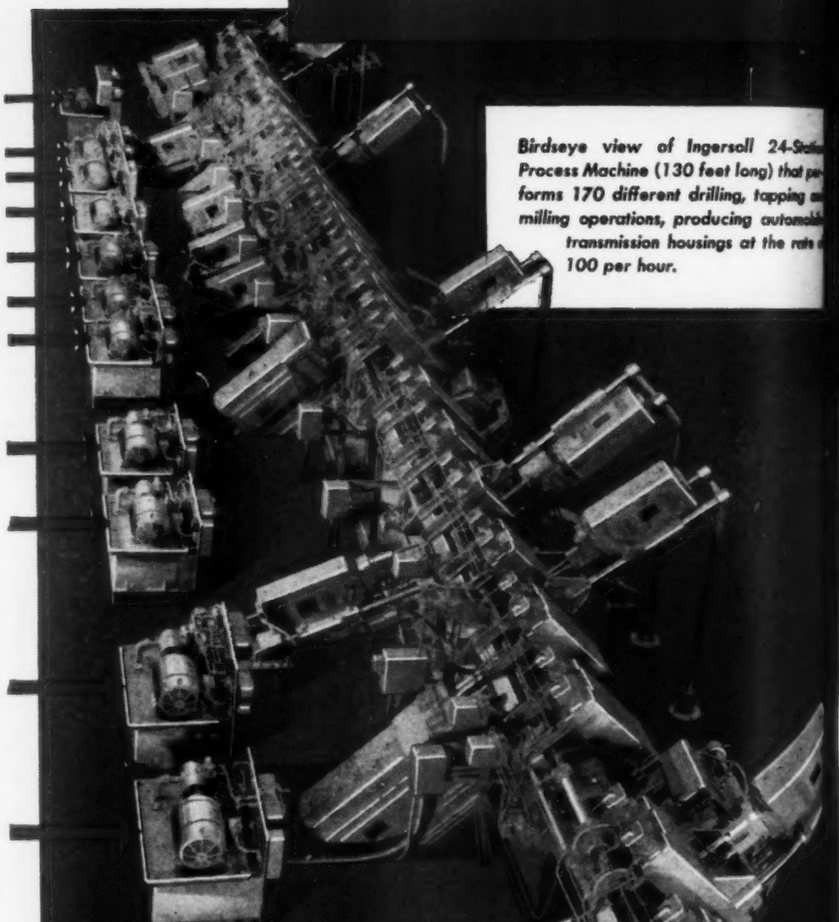

Birdseye view of Ingersoll 24-Station Process Machine (130 feet long) that performs 170 different drilling, tapping and milling operations, producing automobile transmission housings at the rate of 100 per hour.

Write for new Bulletin 46-43 describing Vickers Hydraulic Power Units

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ENGINEERS AND BUILDERS OF OIL HYDRAULIC EQUIPMENT SINCE 1921

Myron Curtis, assistant to the director of engineering, is studying a layout of a new textile machine while project engineer W. A. Hofmann is explaining it to eastern sales manager Gene Gardner, chairman of the Planning Committee.

Finding New Products To Fill Expanded Plants

How the Warner & Swasey Company organized to search out new products that they could manufacture and market and thereby use the expanded plant capacity created by the war.

MOST of the pacemakers among our manufacturing industries expanded their production capacity anywhere from fifty to several hundred percent before or during World War II to meet the pressing needs of war. Buildings were added, new machinery installed, and over-all organizations expanded to cope with the challenges presented by war.

While much of this expansion was financed by government funds, it permanently broadened the scope, horizons, and importance of these companies in their fields of activity and in their plant communities.

When it became obvious that this expansion, whether or not it was government financed, would confront most firms with the necessity for drastic shrinkage just as soon as normal peacetime conditions were restored, the more farsighted organizations started planning to hold their gains by one means or another. This presented special problems to machine tool builders, whose postwar sales outlook was clouded by the certainty of huge machinery surpluses.

As early as January, 1942—less than two months after Pearl Harbor—the management of Warner & Swasey reached the conclusion that

Assembly of pin drafting machines, one of the products resulting from the planned program for finding new machines to manufacture, now occupies a section of the plant built in 1940 for the manufacture of machine tools.

only by promptly launching a long-range program for planning and development work could they avoid a violent postwar collapse. It was then that they began to look for new products through which their skills, know-how, and manufacturing capacity could be profitably employed.

Viewed in retrospect, there was nothing remarkable about the way in which Warner & Swasey attacked their problem. There was, however, one highly significant feature in the method of their procedure. They appointed a Planning Committee, and the sole job of each member of that committee was to find or analyze new products. Much of the success of the Planning Committee is attributed to the fact that every member had nothing to do other than his work on the committee.

Another significant feature was the make-up of the committee. Eugene R. Gardner, eastern sales manager, was made chairman of the committee. The selection of a sales manager to head the work of the committee was for several reasons. First, in every new product venture the sales problems are major ones and they do not lend themselves to complete factual analysis. Judgment plays a predominant part in making sales decisions. Also, the particular sales manager selected had to have the vision and imagination that is so essential to an undertaking of this kind. And finally, Mr. Gardner's activities in selling machine tools had given him a broad insight into both manufacturing and sales problems.

Two members of the Warner & Swasey engineering department, Myron S. Curtis and James R. Longstreet, were put on the committee, full time, to analyze the design, production, and cost problems that would come up. And to assist Mr. Gardner in the many ramifications of the marketing and analytical problems that would arise, Mr. L. M. Cole, manager of the Texas sales division; Mr. L. R. Hawkins, branch manager of the West Coast sales division; and Mr. R. M. Wise, an economist, were appointed on the committee.

The committee was assigned the job of planning and executing the program to find new products. The committee was given carte blanche authority to do as it pleased, and did not require special authorization for any investigation that it saw fit to embark upon.

Several possible directions of procedure were first studied by the committee. A new product might be developed through the findings of basic scientific research. This method was immediately ruled out because it would be too slow in getting results, and Warner & Swasey made no pretense of being a scientific research organization. Secondly, there was the possibility of inventing new products. This also was ruled out as being too slow, uncertain and costly. Finally, there was the plan to seek existing products to manufacture, and this was the plan adopted.

Having arrived at the decision to seek existing products to manufacture, next step was to determine the particular types of products that Warner & Swasey could manufacture with a reasonable chance of success. The answer was arrived at by a process of elimination. For example, it was obvious that many of the com-

panies whose regular line was electric refrigerators, washing machines, or other home equipment items, would have vastly increased production capacities when the war ended. They were not only far better equipped to produce these consumer "hard goods" than was Warner & Swasey, but also had established sales outlets. For these reasons, all consumer goods were ruled out.

Conversely, Warner & Swasey, through their many years in machine tool building, had an organization skilled in the manufacture and marketing of metal working machinery. It was not desirable to confine the new program to the machine tool field because that would merely further concentrate the company's business in the same channel. However, there is considerable similarity between machine tools and most other machinery capital goods, both in the nature of the manufacturing processes involved and the sales problems involved. On this basis, it was decided that the Planning Committee would concentrate its efforts on finding new machinery capital goods.

Having established the general direction in which they were going to seek new products, the Planning Committee then set up the ways and means by which they were going to find these products. Three lines of effort were undertaken. First, all product items that had been submitted for consideration over the recent past were carefully reviewed. Additional products were studied as submitted. And finally, various ways and means were established to widely publicize the fact that Warner & Swasey was in the market for new products. The methods used included advertising and simply "talking it up."

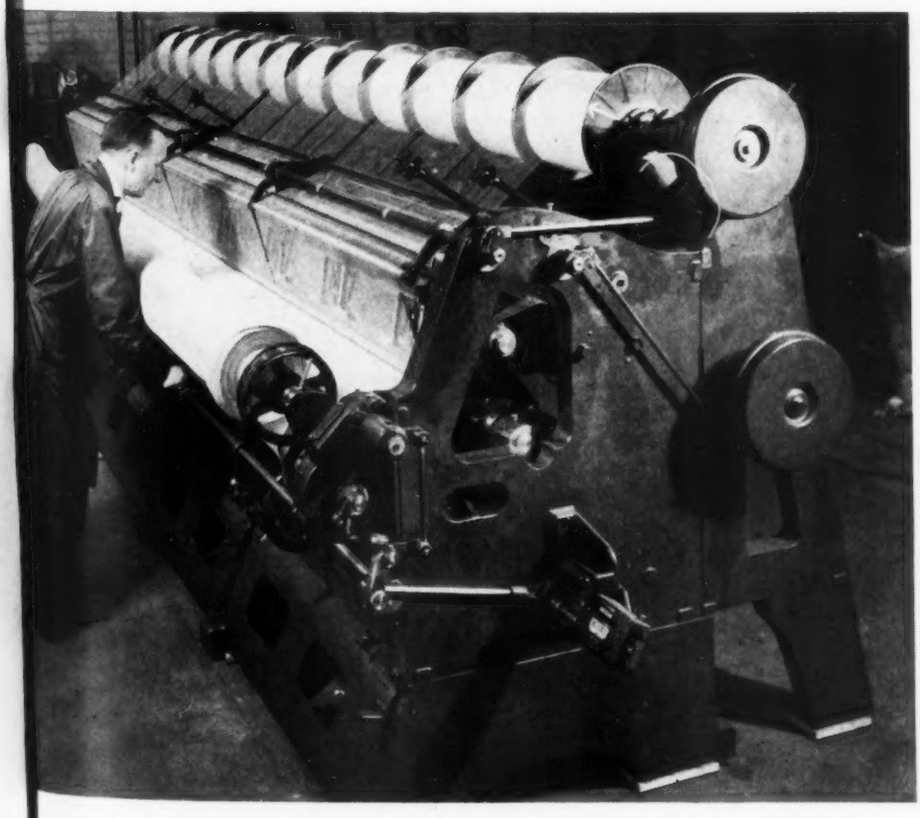
Advertising brought in floods of suggested items, each of which was studied carefully if it held any promise. The more than 3,000 suggestions that came in included electric cow tail holders, mouse traps, numerous infinitely variable speed transmissions, mine locomotives, and, of course, the usual number of perpetual motion devices. All items other than capital goods machinery were called out. The committee selected the "items of merit" that, in their opinion, warranted serious detailed study. If the results were favorable, the item was put before the board of directors to approve initial development and the procurement of manufacturing rights.

One unexpected result of the work of the Planning Committee was that it led into a hunt for specialized engineering design talent within the organization. A textile mill owner, for instance, had a rough idea for a pin drafting machine, used in the manufacture of worsted yarn, and wanted Warner & Swasey to develop and perfect the design. A search in the Warner & Swasey engineering department uncovered a designer who had had extensive experience in the designing of foreign textile machines. The job of developing the pin drafting machine design was undertaken, and the machine is now in production in the Warner & Swasey plant.

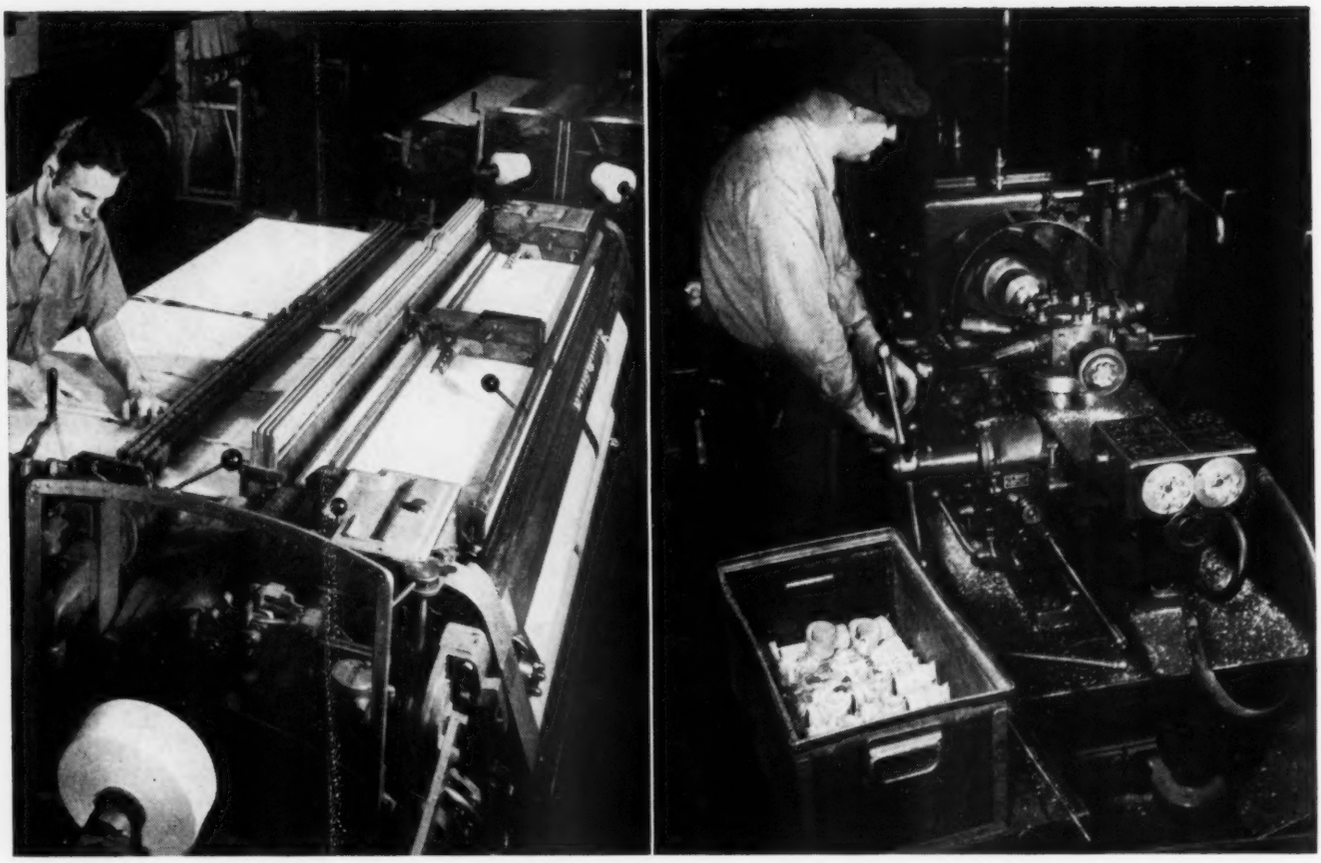
As a direct result of this advertising program, the American Viscose Company—largest American producer of rayon yarn—called Warner & Swasey in to discuss a new rayon knitting machine. They had the design, and were looking for a manufacturer whose familiarity with problems in the volume production of precision machinery and whose financial responsibility would justify the award of a contract for exclusive manufacture. When Warner & Swasey turned its engineers loose on the study of the machine, their analysis of the production problems disclosed several opportunities to cut costs by certain design changes, which in no way affected the mechanical aspects of the equipment. These were an important consideration in the final determination of a fixed price contract.

When they got into the production of this Aveco tricot knitting machine, Warner & Swasey introduced further economies by substituting a welded base for cast iron, automotive type bearings for the journal type originally specified, and a number of machined parts in place of more expensive castings at many points in the machine.

One exceptionally interesting piece of equipment that was added to the Warner & Swasey products list is the Gradall, invented by Mr. Ray Ferwerda, an experienced Cleveland contractor. The inventor had built a number of units on a "tailor made" basis. He was seeking a company that might redesign it for quantity production, manufacture the ma-



One of the first Aveco knitting machines to come off the production line at the Warner & Swasey plant. Later the cast frames and base were replaced by weldments. The machine produces finished rayon jersey fabric up to 14 ft. wide.



License to make and sell a revolutionary Swiss design of weaving machine (left) was obtained and the machine was redesigned for American use and economical manufacture. New machine tools such as the Electro-Cycle automatic turret lathe (right) were developed to broaden the Warner & Swasey line of machine tools.

chine, and market it. A Cleveland man to whom the inventor told his story had heard that Warner & Swasey was actively seeking new products. He advised Mr. Ferwerda to see Warner & Swasey. The final result was that Warner & Swasey redesigned the machine and is now producing it under a manufacturing license to make and sell the machine on a royalty basis.

Here was a new product that manufacturing-wise is considerably different from machine tools. Most of the elements of the machine are plate or structural members involving an extensive amount of welded assembly. It requires only little of the precision machining that characterizes machine tools, and presents special marketing problems that have no relation to machine tools. To resolve these differences, the Gradall machines are built in a separate department and are sold through well established construction machinery dealers.

There were many excellent by-products that came out of the work of the Planning Committee. Salesmen and engineers, especially the committee members, found it necessary to visit many types of plants which they had never seen before, such as knitting mills and other textile plants. Thus, Warner & Swasey engineers were thrown into many new problems in fields of design with which they had no previous acquaintance.

The manufacturing division was given new kinds of production processes to master, as in the manufacture of the structural elements of the Gradall and in the production of the automic features of knitting machines. They were also compelled by circumstances to study in detail the marketing methods in various unfamiliar fields. Thus, the whole Warner & Swasey organization received a broad education, new experiences, and made new discoveries.

With the immediate objectives largely accomplished, the Planning Committee now has the added assignment to study more intensively than ever before, the redesign of the Warner & Swasey turret lathes. Meanwhile, they have supplemented their established line of turret lathes by the addition of three new machine tools which were developed or acquired during the war, and have since been gaining steadily a wider acceptance in the machine tool market. The three new machine tools are:

1. The Warner & Swasey Multiple Spindle Automatic, which is designed for the making of parts, by the turning process, in repetitive quantities, effecting for such purposes cost reductions above those that can be accomplished with the use of regular turret lathes.

2. The Warner & Swasey Electro-Cycle Turret Lathe, developed for the machining of parts made from brass, aluminum and other non-ferrous materials.

3. The Warner & Swasey Tapping and Threading Machine, designed to perform work of this type to extreme accuracy.

The experiences gained in the studies of many fields of endeavor and studies of the manufacturing equipment required in the production of a great variety of products has been found invaluable for this continuing program of product design improvement.

FLUID FLOW FRICTION FACTORS

For Pipes, Valves and Fittings

V. L. STREETER
Armour Research Foundation of Illinois Institute of Technology

Methods of determining losses in fluids flowing through pipes and fittings are discussed, as well as methods of expressing these losses in a form useful for design purposes. Losses in valves and fittings are analyzed, and some new experimental loss data on valves are presented.

WHEN FLUID is conveyed through conduits, ports, valves or other fittings a loss in mechanical energy per unit weight of fluid occurs. The lost energy is connected into heat through the action of turbulence in causing viscous shearing of the fluid. As heat the energy is usually of no value in a liquid circuit, it may be detrimental in decreasing the viscosity and thereby causing additional slip in fluid motors and pumps. Furthermore, the losses in conduits must be kept at a reasonably low value to maintain high overall efficiencies for any fluid system.

The energy loss occurring in a particular fitting is usually defined as the additional loss imposed on the system when the fitting is in place over that when straight pipe of the same length as the fitting replaces the fitting. A given fitting may have many different energy losses for a given flow of a specified fluid, depending upon the upstream and downstream conduit arrangements. As there is no limit to all these possibilities, the losses cannot be predicted closely unless a test has been run on a similar system. In general, fitting losses are determined with straight conduit both upstream and downstream for at least 30 diameters. The experimental results from these tests give little information as to the loss caused by the fitting when it is preceded or followed closely by other fittings.

Loss occurs because of disarrangement of the velocity distribution in the conduit. This disarrangement produces additional turbulence that is dissipated by viscous shear into heat through and downstream from the fitting. To determine losses experimentally a testing system is set up in which fluid may be passed through conduits at a variable measured rate. The Armour Research Foundation set-up is shown in Fig. 1. Usually the straight sections both upstream and downstream from the fitting contain piezometer rings, each of which determine the static pressure in the pipe at a cross-section. The piezometer openings are small holes drilled radially through the pipe wall, and smoothed on the inside of the pipe surface to avoid projections that would disturb the flow.

With a piece of straight pipe inserted in place of the fitting that is to be tested, the set-up is calibrated. Energy losses resulting from conduit friction are determined for several discharges. In Fig. 2 the pressure in the pipe is plotted for one discharge. The solid line shows the pressure

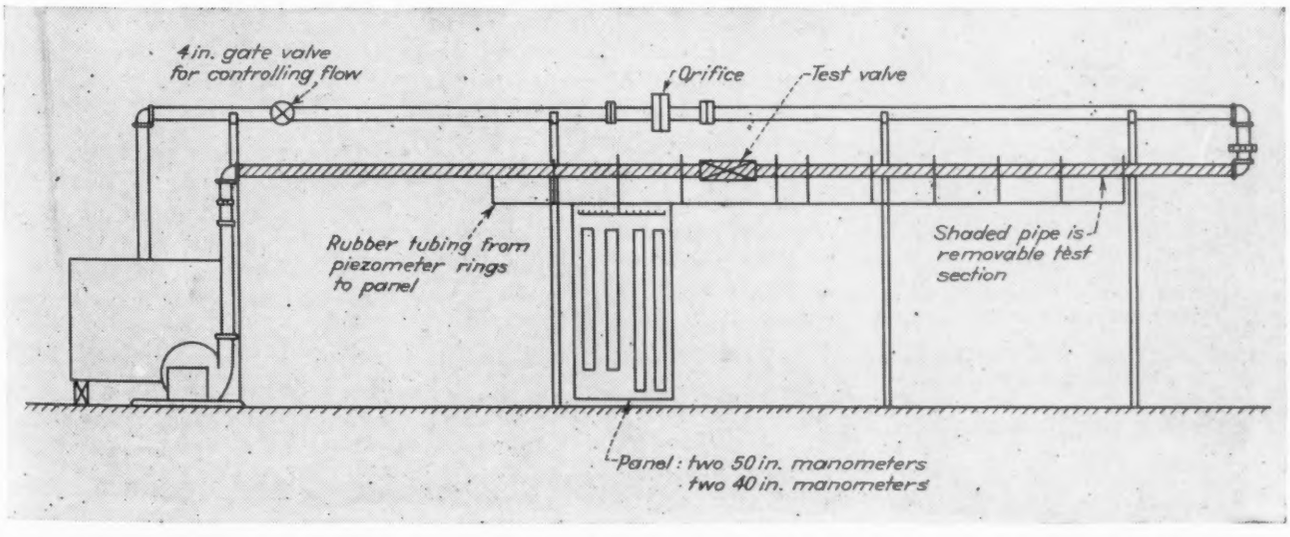

4 in. gate valve for controlling flow
Orifice
Test valve
Rubber tubing from piezometer rings to panel
Shaded pipe is removable test section
Panel: two 50 in. manometers two 40 in. manometers

Fig. 1—Arrangement of apparatus at Armour Research Foundation for determining loss in fluid flow through valves.

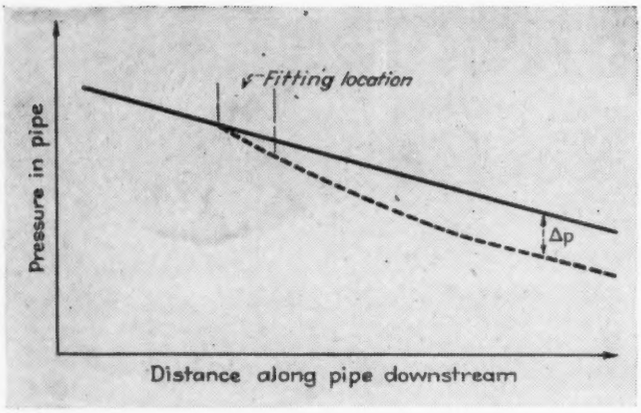


Fig. 2—Pressure in conduit with fitting for a given discharge.

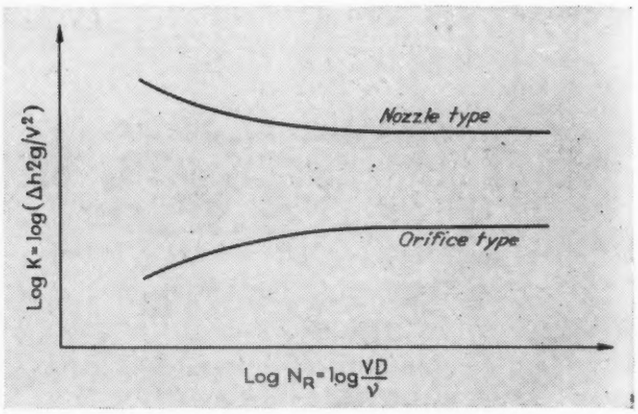


Fig. 3—Log-log plot of fitting K factor and Reynolds number.

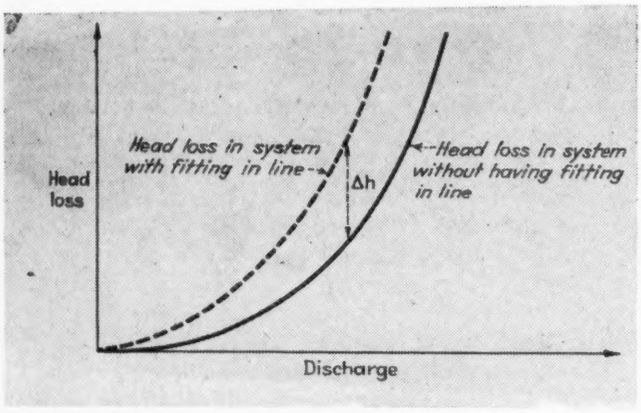


Fig. 4—Loss in a system with and without a fitting in the line.

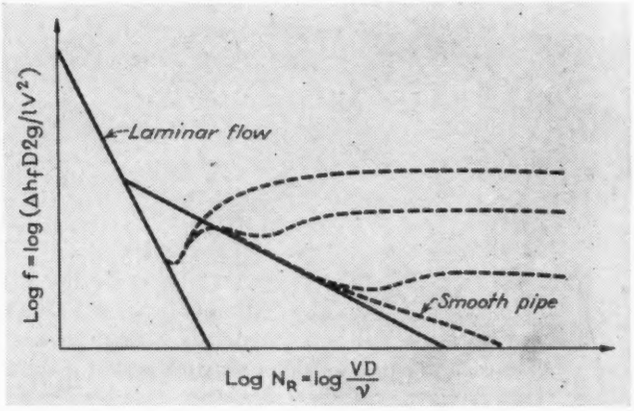


Fig. 5—Friction factor and Reynolds number for pipe friction.

with straight conduit inserted in place of the fitting, and the dotted line indicates the pressure when the fitting is inserted in the system. At some distance downstream from the fitting these pressure curves again become parallel. When straight conduit precedes and follows the fitting, the vertical distance between these curves is proportional to the energy loss in the fitting for this discharge.

Methods of Expressing Flow Losses

The Fanning formula is commonly utilized to express pipe friction losses. Where

f = friction factor, dimensionless
l = pipe length, ft.
D = inside diameter of pipe, ft.
v = average velocity of flow, ft. per sec.
g = acceleration of gravity, ft. per sec. per sec.
Δh_f = head loss resulting from friction in the length l, feet of fluid flowing

The Fanning formula is

$$\Delta h_f = f \frac{l}{D} \frac{v^2}{2g}$$

For fully developed turbulent flow (high Reynolds numbers) fitting losses vary as the square of the velocity. Where Δh is the head loss, expressed in feet of fluid flowing, caused by a fitting, the dimensionless loss factor K for a fitting may be defined by the relation

$$\Delta h = K \frac{v^2}{2g}$$

The loss factor K may be determined from Fig. 2, by dividing ΔP by $wv^2/2g$, where w is the unit weight of fluid. K may be thought of as the fitting loss, expressed in velocity heads. Another method of expressing fitting losses is by means of the "equivalent length" of straight pipe that would produce the same head loss when inserted in the system as that caused by the fitting.

Where

l_e = equivalent length of straight pipe, ft.

the Fanning formula becomes

$$\Delta h_f = f \frac{l_e}{D} \frac{v^2}{2g} = K \frac{v^2}{2g}$$

From this relation it is seen that

$$l_e = K D/f$$

This relation shows that the equivalent length is not constant, even for fully developed turbulent flow, because f depends upon the interior roughness of pipe. The loss factor K is negligibly affected by variations in pipe roughness. K is not a constant for all values of the velocity; in fact, it is a function of Reynolds number N_R.

Where

ν = the kinematic viscosity, sq. ft. per sec.

$$N_R = \frac{vD}{\nu}$$

It is customary to plot K against N_R on a log-log chart. When this is done, one or the other of the two types of curve shown in Fig. 3 results. When the fitting is such that a considerable contraction takes place, as with a sharp-edged orifice, the concave-upward curve results; while a change of the type caused by a Venturi-meter or nozzle causes the concave downward type of plot.

To obtain K from the experimental data for a wide range of discharges, and hence a wide range of Reynolds

number, an auxiliary plot similar to that shown in Fig. 4 can be utilized.

The K factor is the best method of expressing losses, as it is constant for fully developed turbulence. For most uses this constant value will suffice. The equivalent length of pipe l_e can then easily be obtained for particular application to a given pipe system when the friction factor f is known. In many problems f is unknown because the discharge is unknown, and the equivalent pipe method of computing becomes cumbersome. The equivalent pipe method is convenient: (1) Where accuracy is not required or is rendered impossible by fittings following each other in close succession; and (2) where a large amount of work is to be done with a particular kind of pipe and a limited Reynolds number range.

Pipe Friction

Using the Fanning formula for pipes, the friction factor is determined experimentally and is customarily plotted as in Fig. 5. The straight line marked laminar flow can be derived from Newton's law of viscosity and does not need to be obtained experimentally. For incompressible fluids the fluid friction loss is independent of the pressure. For laminar, or "straight-line" flow, where the fluid particles move parallel to the axis of the conduit, the friction loss is also independent of the roughness of conduit walls.

At values of Reynolds number greater than 2,000 the flow usually becomes turbulent in conduit systems, but may remain laminar for Reynolds numbers as high as 40,000 in laboratory set-ups where great precautions are taken to avoid disturbances in the fluid. For rough pipes in turbulent flow the lines in Fig. 5 become horizontal for large Reynolds numbers, thus indicating that in this range a change in viscosity does not affect the losses. This condition is referred to as fully developed turbulent flow, where the head loss is proportional to the square of the velocity. For smooth pipes the effect of viscosity remains evident at large Reynolds numbers. Wavy wall roughness also continues to show a viscous effect for high Reynolds numbers.

Valves and Fittings

Valves and fittings disrupt the normal pattern of turbulent flow in a conduit so that the losses are proportional to the square of the velocity for large Reynolds numbers. Table I gives a list of minor loss K factors as presented in the Crane Company, Technical Paper No. 409 "Flow of Fluids." In Table II are given K factors as determined at Armour Research Foundation for plug and diaphragm types of valves.

In tests of a high pressure valve of nominal 1 in. size using double extra strong pipe and standard pipe, the data showed that for the full open position the loss was 9 percent greater with the standard pipe.

As valve and fitting losses in general cannot be predicted within 10 percent, the size of approach pipe appears to be of secondary importance in determining the fitting loss.

The losses listed in Tables I and II are for straight conduit both upstream and downstream from the valve or fitting. What is the loss when this is not the case? Our present knowledge of losses is inadequate in determining these special cases except by a test of a similar set-up.

Table I—Representative K Values for Valves and Fittings

Type	K	Authority
Globe valve	10.0	Crane tests
Angle valve	5.0	Crane tests
Swing check valve, fully open	2.5	Crane tests
Close return bend	2.2	
Standard tee	1.8	Giesecke and Badgett
Standard elbow	0.9	Giesecke and Badgett
Medium sweep elbow	0.75	Crane tests
Long sweep elbow	0.60	Bulletin No. 2712 Univ. of Texas
45 deg. elbow	0.42	Bulletin No. 2712 Univ. of Texas
Gate valve		
fully open	0.19	Bulletin No. 252 Univ. of Wisc.
1/4 closed	1.15	Bulletin No. 252 Univ. of Wisc.
1/2 closed	5.6	Bulletin No. 252 Univ. of Wisc.
3/4 closed	24.0	Bulletin No. 252 Univ. of Wisc.

Table II—K Values for Plug and Diaphragm Valves

Losses listed are based on the velocity in standard pipe, except for the 600 lb. per sq. in. globe valve in which extra strong pipe is used

Type	K
Plug globe, or stop valve	
600 lb. per sq. in. fully open	4.0
3/4 open	4.6
1/2 open	6.4
1/4 open	780.
Diaphragm valve	
fully open	2.3
3/4 open	2.6
1/2 open	4.3
1/4 open	21
Plug valve	
(1/4 turn from closed to full open)	
fully open	0.77
99 percent open	0.86
98 percent open	0.95
95 percent open	1.45
90 percent open	2.86
80 percent open	9.6
70 percent open	28.0

Conclusion

Published data aid in making rough estimates of the loss resulting from pipe friction, valves and fittings. It is questionable that losses can be estimated closer than 10 percent in a system having many fittings. As the possible number of combinations of systems is unlimited, it is out of the question to expect to amass sufficient experimental data in advance so that any given system can be analyzed. More experimental work is needed, as it will improve accuracy, but the ultimate solution of the problem must be found by basic research in the development, transmission and decay of turbulence in a fluid. When turbulence is understood, flow lines may be designed to reduce its effect to a minimum, thereby saving much in power lost in conveying fluids through conduits.

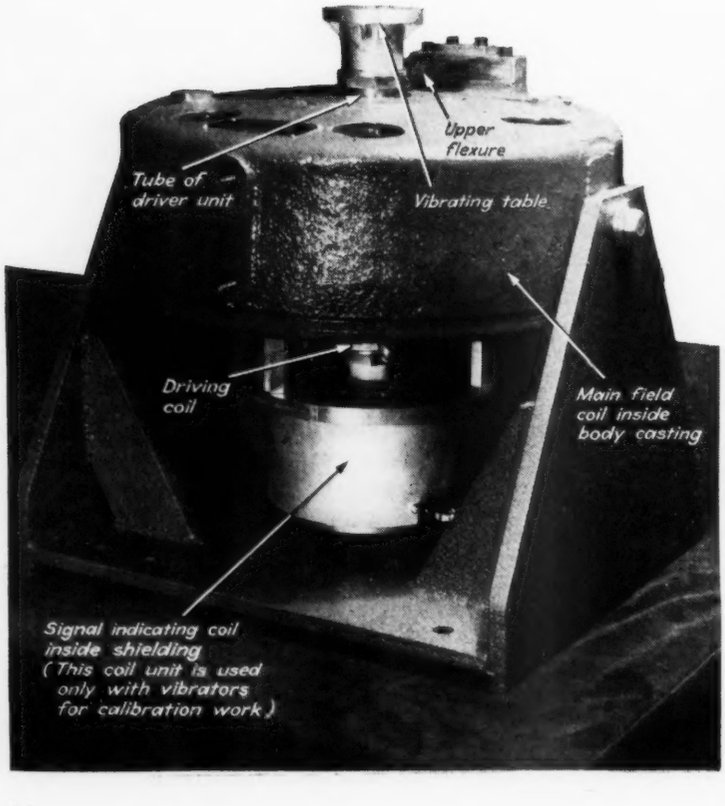
Such fundamental work is under way at many research centers. The Armour Research Foundation and Graduate School of Illinois Institute of Technology are now undertaking a long range project that will contribute to knowledge of fluid losses.

PRODUCT DESIGNS

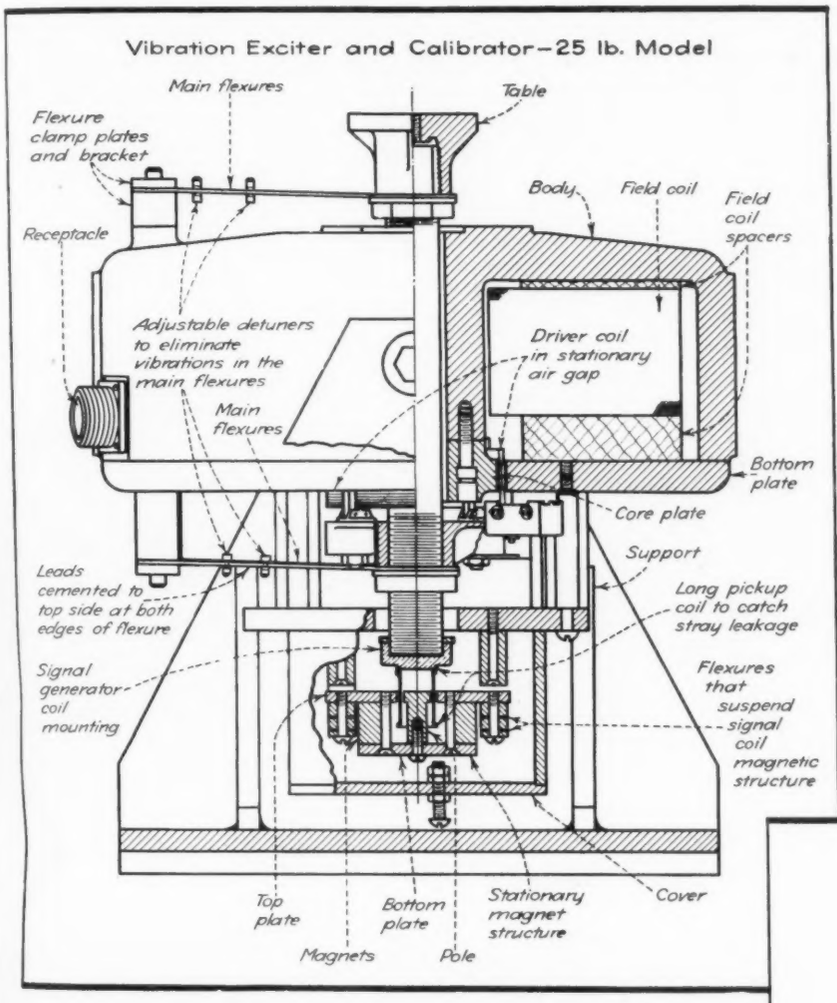
Typical Setup Using the Vibration Exciter to Calibrate a Vibration Pickup

Electro-Magnetic

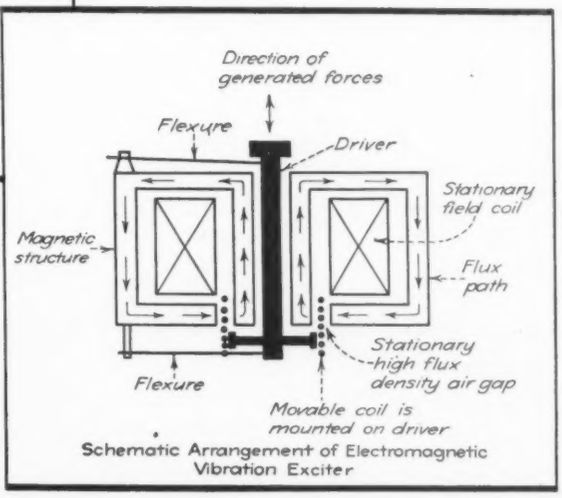
Vibration Exciter and Calibrator


Upper flexure
Tube of driver unit
Vibrating table
Driving coil
Main field coil inside body casting
Signal indicating coil inside shielding (This coil unit is used only with vibrators for calibration work.)

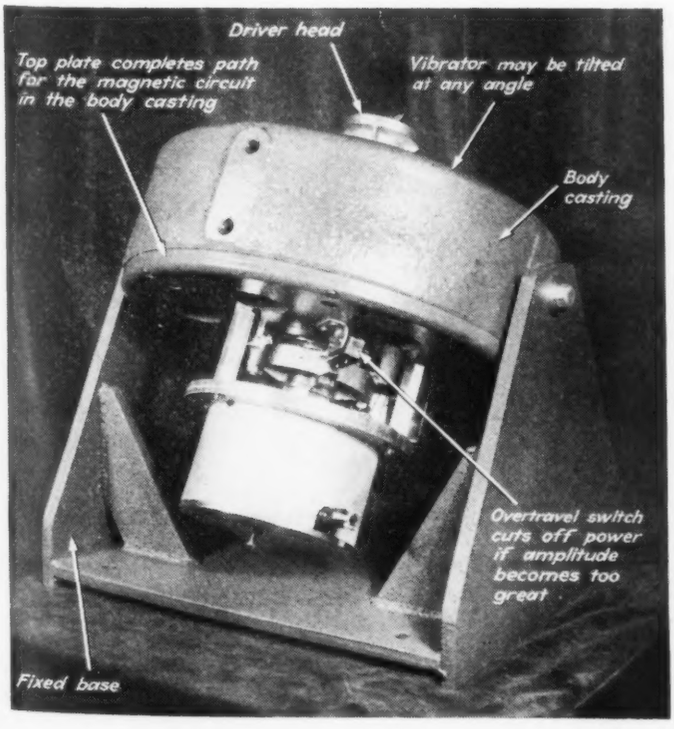
DYNAMIC stress analysis using a vibration exciter is a useful testing technique for checking structural performance. The method is discussed on page 115. Vibration exciters are not new, but most of these devices have been mechanical in design, employing inertia unbalances for force generation. They have been handicapped by their size, the weight added to the system under test, lack of frequency range and force adjustment while operating. By electromagnetically driving a small lightweight table, or driving head, that can be easily attached to any point and in any plane of a structure, the MB Manufacturing Co., Inc., New Haven, Conn., has achieved a versatile and sturdy exciter covering low and high frequency ranges. These exciters are rated at 5, 25 and 200 lb. minimum vector peak force available continuously to 500 cps. at least. The 25 pound model, illustrated, will generate peak-to-peak displacements up to ½ inch. This model carries a signal-generating coil to adapt the exciter for calibration of vibration pickups. Operating range is 4—500 cycles as a calibrator; up to 70,000 cycles as a vibrator. The force, limited only by the power supply, falls off gradually above 500 cps. Greater power supplies will produce 100 lb. and larger forces on a short time rating basis, limited by heat generated in the driver coil.

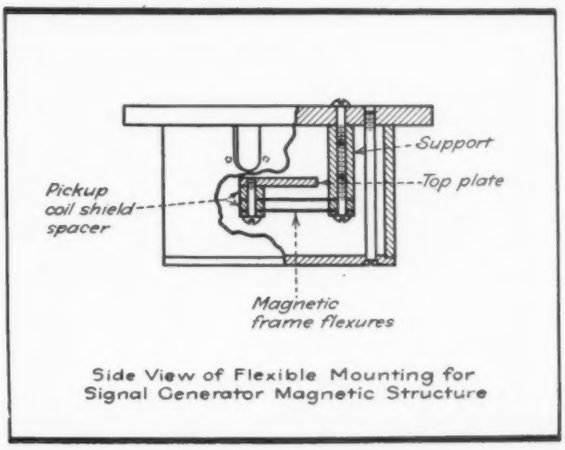


Vibration exciter comprises a stationary, trunnion-mounted, magnetic structure, inclosing a field coil and a moving driver assembly. Driver assembly of table, tube and coil are supported from the magnetic structure by flexures that allow the driver to move only along its own axis. Power source includes oscillator, power amplifier for the a-c driver coil and selenium rectifier for d-c field coil. All operate from house current. The driver coil is mounted in the high-flux-density air gap in the stationary magnetic structure. The coil derives its force when carrying a current, in accordance with the laws of electromagnetic induction.



Schematic Arrangement of Electromagnetic Vibration Exciter

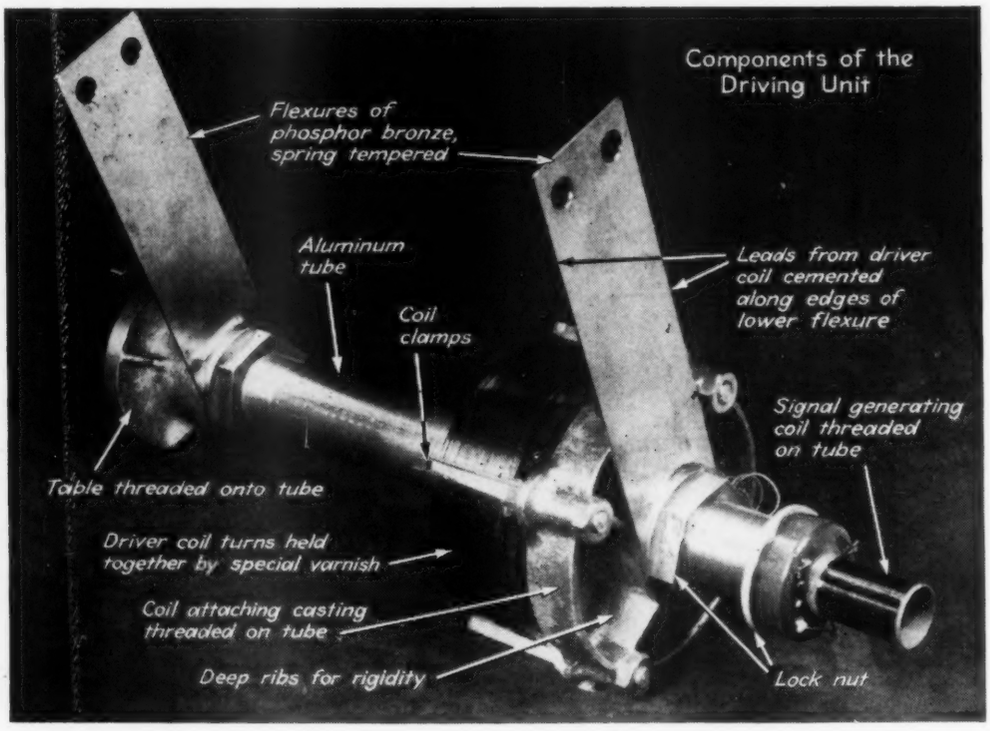

Driver head
Top plate completes path for the magnetic circuit in the body casting
Vibrator may be tilted at any angle
Body casting
Overtravel switch cuts off power if amplitude becomes too great
Fixed base



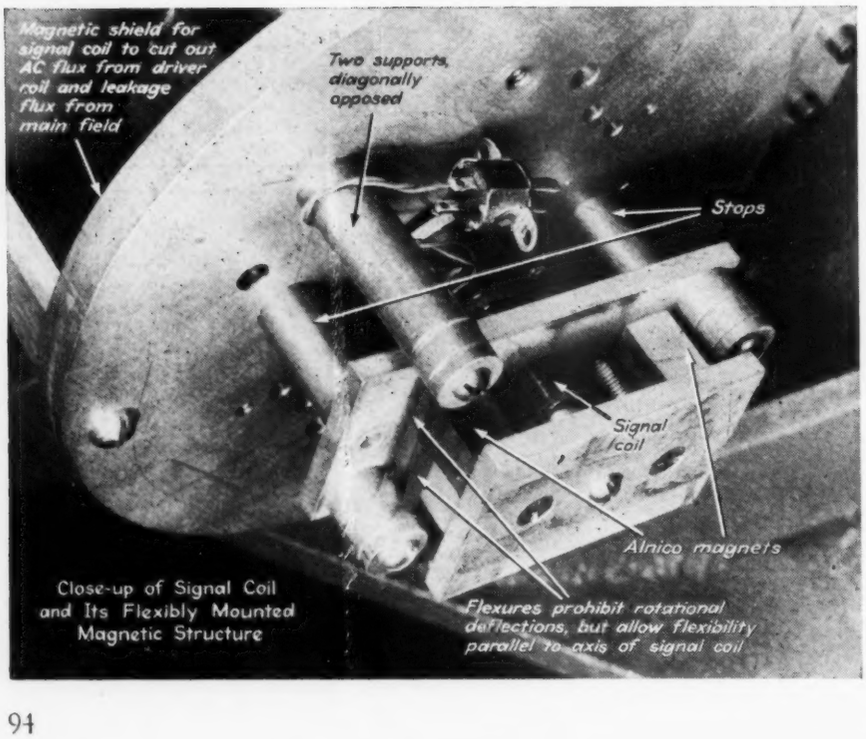
Side View of Flexible Mounting for Signal Generator Magnetic Structure

(continued on next page)

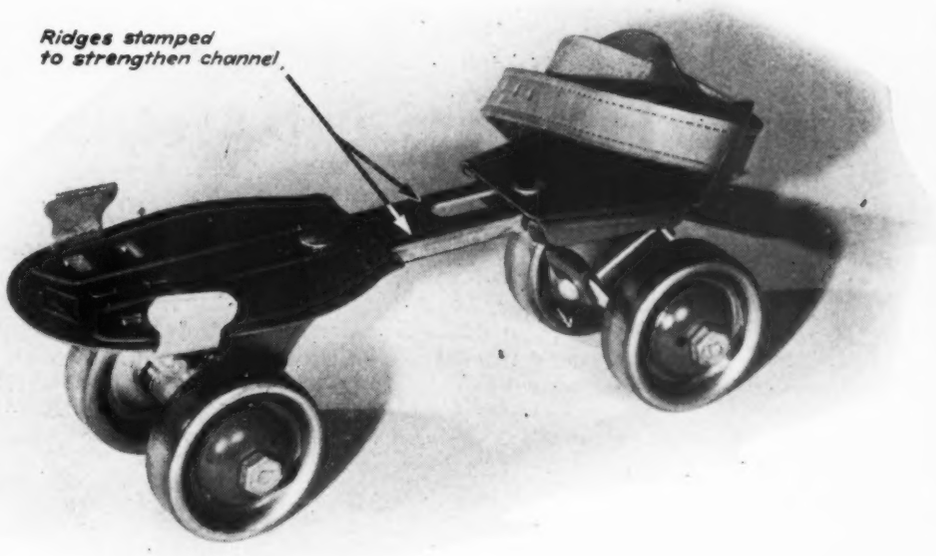
Vibration Calibrator (continued)


Components of the Driving Unit
Flexures of phosphor bronze, spring tempered
Aluminum tube
Coil clamps
Leads from driver coil cemented along edges of lower flexure
Signal generating coil threaded on tube
Table threaded onto tube
Driver coil turns held together by special varnish
Coil attaching casting threaded on tube
Deep ribs for rigidity
Lock nut

Design problem is to keep resonances out of the system by making the driver unit as **rigid and compact as possible along axis of** movement. Shortening the tube raises the natural frequency and reduces deflection for a given load. Tube diameter is large for stiffness considerations, but size must be compromised with need for light weight in tube, coil and coil mounting casting. **Flexures** must be stiff in a plane normal to driver motion but flexible along the line of driver action. Flatmembers have been found most satisfactory for simplicity, stress and **resonance considerations.** Flexure thickness is limited by stress in outermost fibers. Flexures must be long so that arc motion at the tip will approximate a straight line, but length must be compromised to minimize resonance in the flexure. Detuners are positioned along the flexures to damp flexure vibrations if and as required.

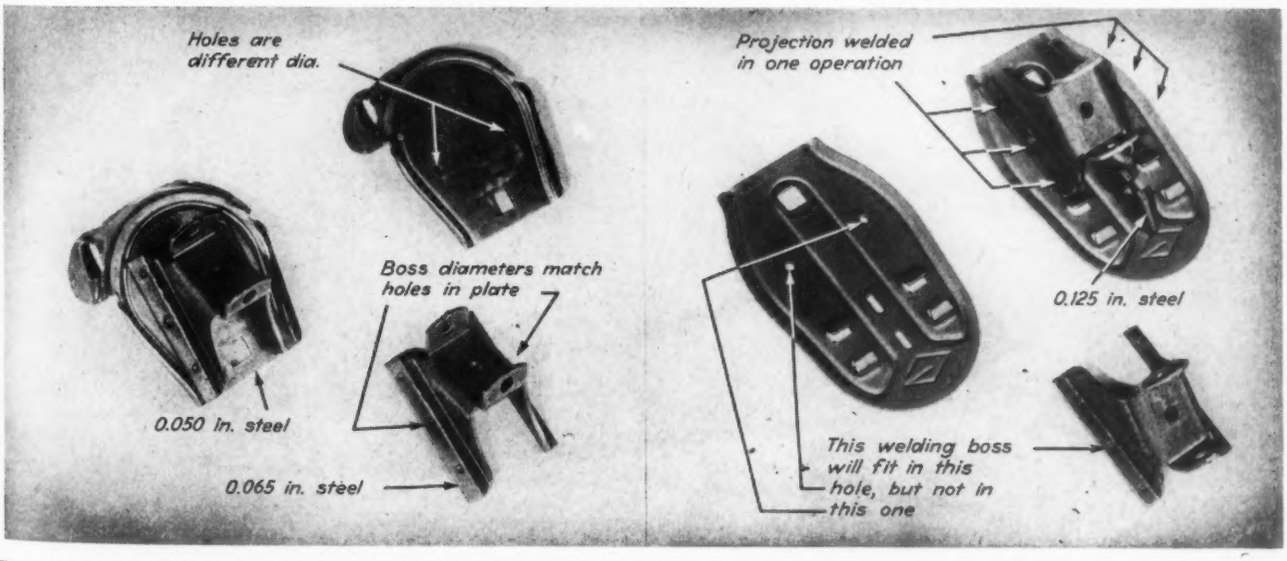

Magnetic shield for signal coil to cut out AC flux from driver coil and leakage flux from main field
Two supports, diagonally opposed
Stops
Signal coil
Alnico magnets
Flexures prohibit rotational deflections, but allow flexibility parallel to axis of signal coil
Close-up of Signal Coil and Its Flexibly Mounted Magnetic Structure

To obtain the maximum amount of copper in a minimum air gap width, the driver coil is not wound on a coil form. The coil is supported by a clamp arrangement and a special baking varnish gives high strength and rigidity. Calibrator models have a signal generating coil mounted on the bottom end of the driver. A magnetic structure is mounted on the housing to furnish a stationary magnetic field for the signal coil. This magnetic structure is flexibly mounted, so that the signals will reflect driver motion and not resonance within the housing. Electrical and mechanical properties of signal generator give constant generator sensitivity proportional to velocity through the frequency range from 0 to 500 cps.



Redesign For Projection Welding Speeds Stamping Assembly

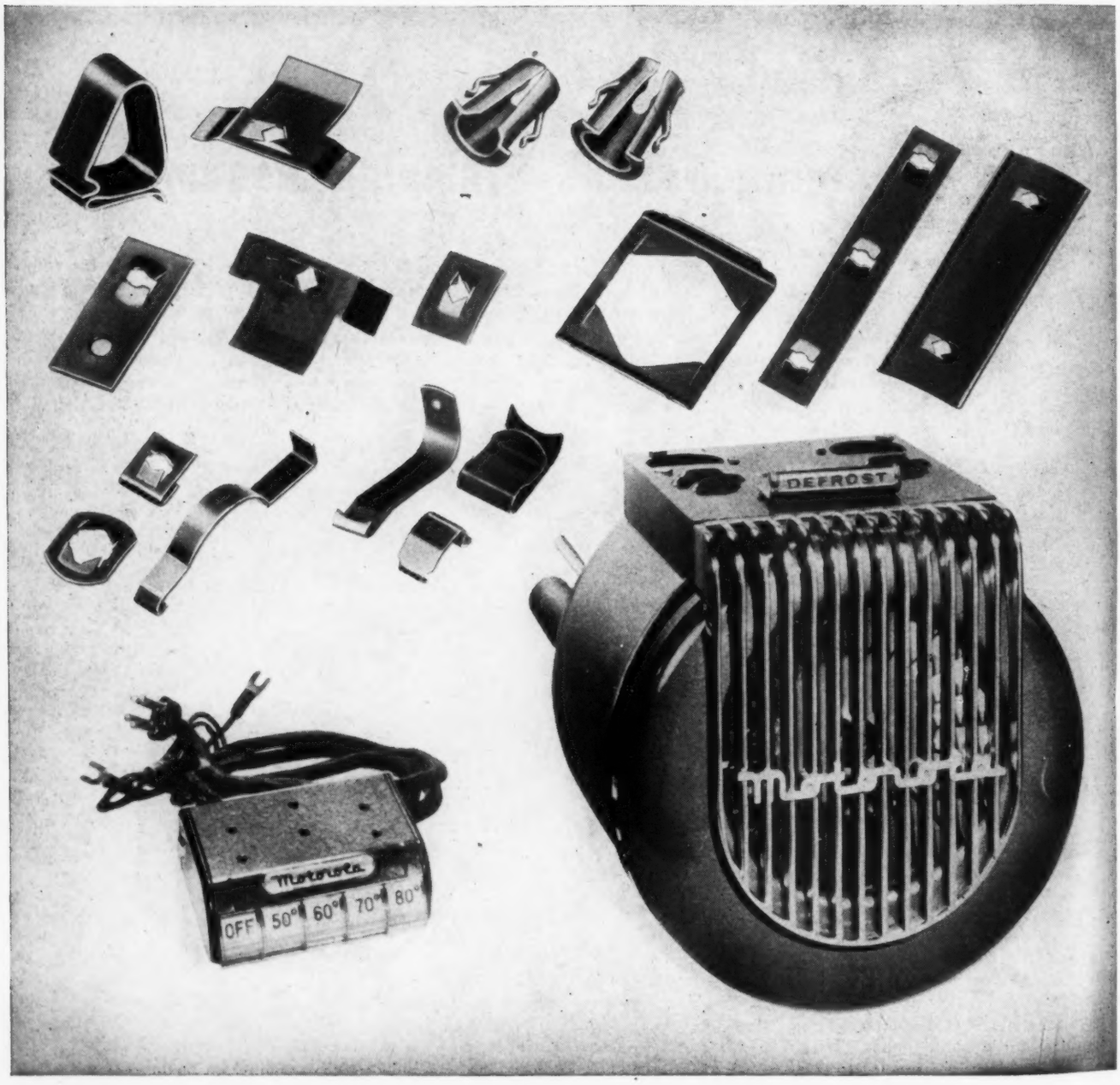
BOSSES for projection welding serve as locator pins to position the parts in the welding of this redesigned roller skate. Formerly the stamped brackets of the skate were joined to heel and toe plates with four or more spot welds. Complicated jigging was needed to insure that plate and bracket were aligned properly. Assembly time was cut one-third when the stampings were redesigned so that two of the projection welding bosses located the pieces in such a manner that they can be assembled only in correct orientation. In addition to reducing the assembly time, this redesign enables inexperienced welders to do the work. The skates are made by George K. Garrett Company, Inc., Philadelphia.

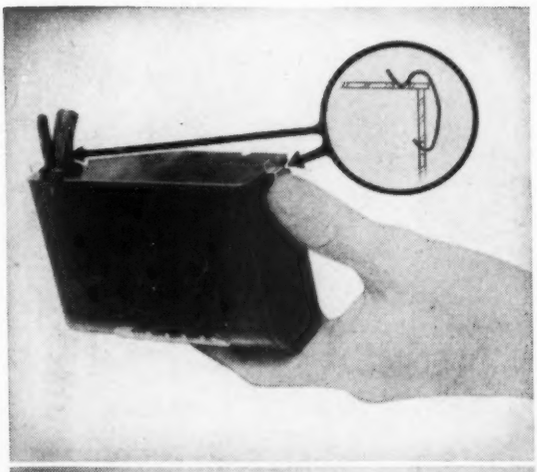


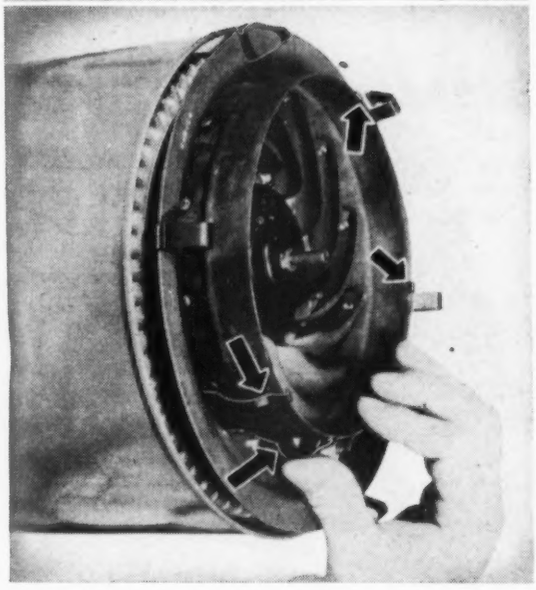
Two Locating holes of different diameters are pierced in both toe and heel plates. Two of the six projection weld bosses in each bracket are punched to match the respective hole diameters in the plate. Thus when the parts are assembled for welding, the bosses fit into the holes to establish bracket location. Discrepancies in diameter between holes and bosses prevent the bracket from fitting on the plate backward. The six projection welds on each plate are made in one operation. The extension bar has been strengthened from its original form as a plain channel, 0.075 in. thick, by stamping two ridges parallel to the long sides and then case hardening all surfaces of the bar.

Special Fasteners Cut Assembly Time

FORTY-FOUR sheet metal fasteners of 14 different types are used to assemble the Motorola Gasoline Car Heater, built by Galvin Mfg. Corporation, Chicago. This type of assembly was planned while the design was in the mock-up stage, so that with slight changes in heater parts and by provision of unthreaded, integrally molded studs, standard fasteners were adapted as widely as possible. Rapid assembly results because many parts are snapped into position under the spring tension of clips, others by pressing fasteners over the integral studs. Compactness and ease of disassembly are retained. The spring-tension loading of this type of fastener prevents vibration loosening. The SPEED NUT fasteners, made by Tinnerman Products Inc., Cleveland, need no lock washers or spanner washers and have no threads to clog with paint or rust to the bolt.

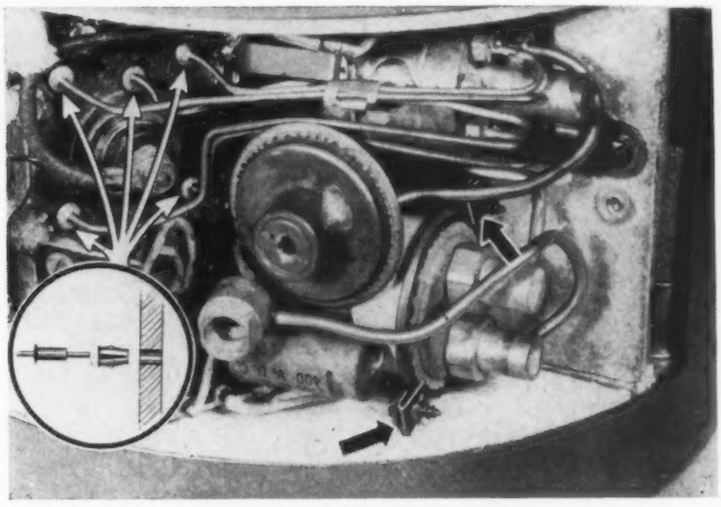




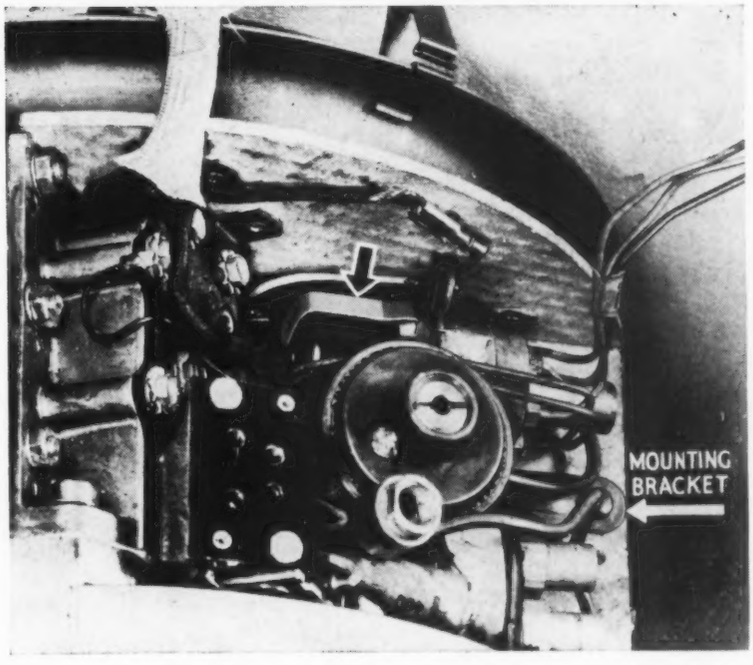


Two spring clips hold the backplate on the control head housing. They are snapped easily into locked or unlocked position. Similarly four clips are hooked under sheared bridges on the deflector plate and snapped over the edge of the directional air flow ring to lock it in place.

Insert shows how fuel lines are attached to burner block. Eyelets, soldered to lines, are pressed into tubular clips installed in the block. Annular groove at end of clip has enough tension to hold the eyelet securely, yet permit easy removal of fuel line. Black arrow points to "J" type fastener that holds the selector to a bracket. Short lower leg acts as spacer so that the nut is flush with the selector flange.



Mounting bracket for filter selector and fuel pump. They are attached before installing in the heater as a unit. Forked leg of bracket is pressed under spring arm of nut and straddles the plain rivet that is held to the asbestos panel by this nut. Other bracket leg is attached by tightening a screw into a similar fastener (not illustrated) that is riveted to the panel.

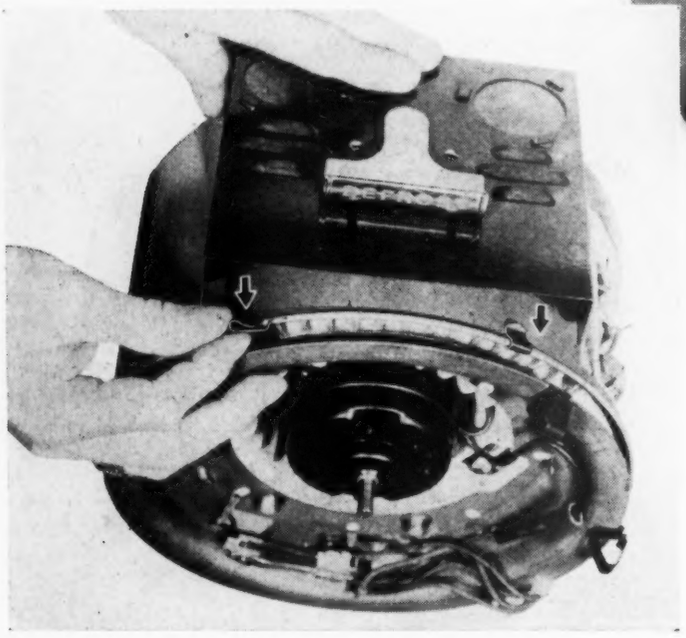


Plastic bellows housing is installed with two spring-latch type clips. The clips hook into a sheared louver in the metal base and the boss snaps into an annular groove in top of housing. For porcelain, plastic or glass parts this type of fastener is advantageous because it has adequate tension yet is resilient enough to prevent cracking.

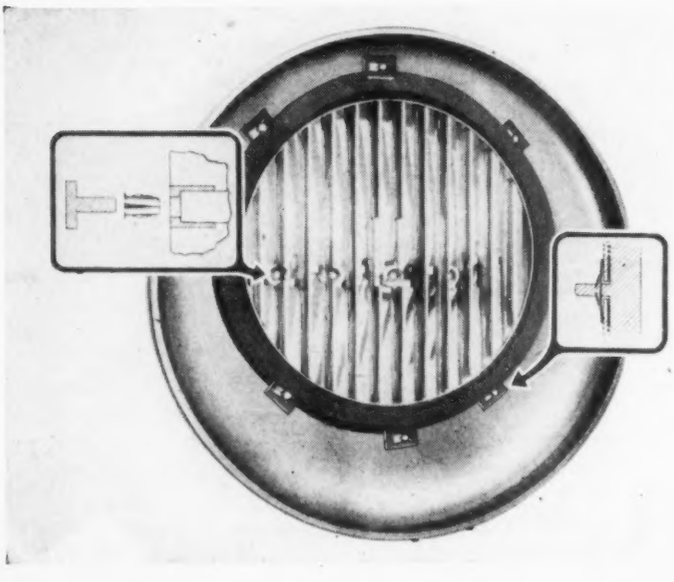
(continued on next page)

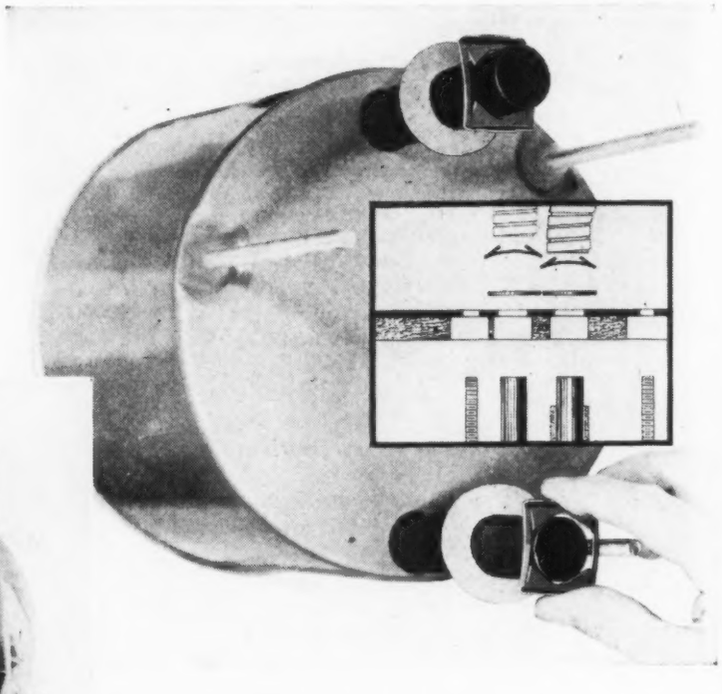
Fasteners (continued)

"U" shaped clips are pressed over rolled edge of heater shell for attachment of defroster housing. Front end of housing is pushed under spring arms of clips and two screws at the rear complete attachment.

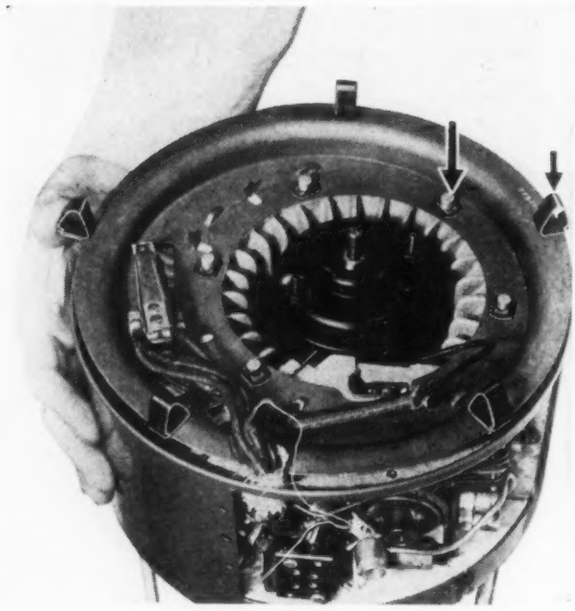


Die cast fan grille is attached by pressing push-on type Speed Nuts over integrally molded studs. Tubular clips hold name plate overlay to grille. Clips are pressed into holes in the grille, then overlay studs are pressed to lock in the clips.



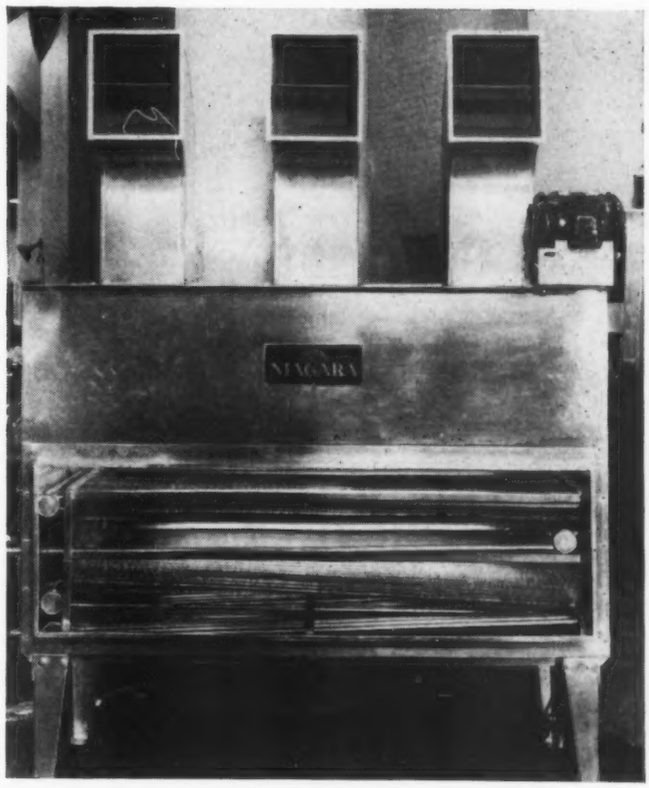


After the heater is mounted to dash panel, large push-on fasteners are pressed over intake and exhaust tubes. They hold asbestos washers against the back of the dash.

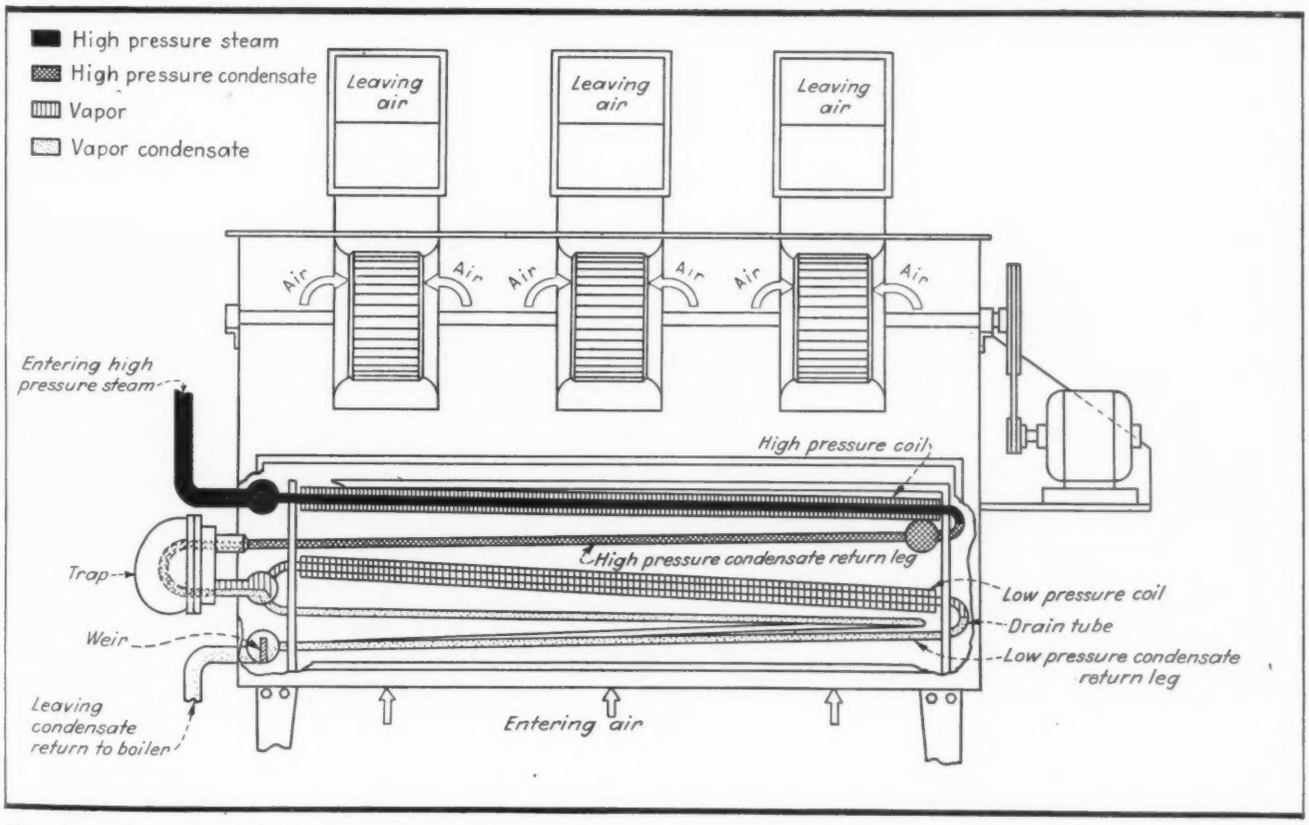


Heat deflector plate is attached by standard push-on type Speed Nuts. They are pressed over integral studs of the heat exchanger. On the rim of the deflector plate are self-retained spring clips. They hold the front cover on final assembly.

Two-Stage Condensation Improves Steam Heater Performance

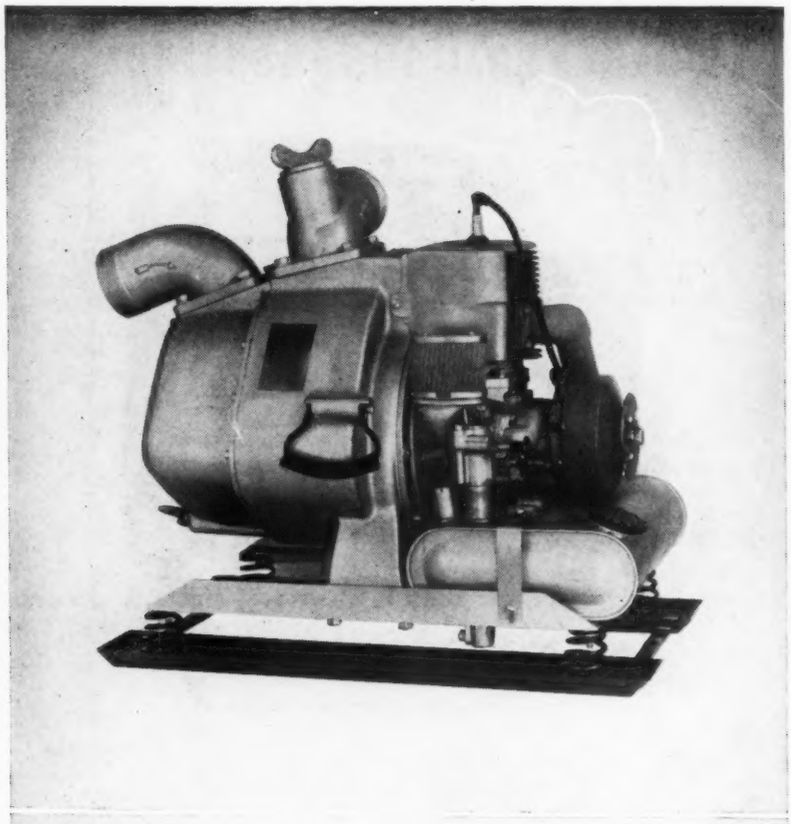


A DUAL COIL SYSTEM to utilize more completely the latent and sensible heat in high pressure steam is the heart of the Niagara High Pressure Fan Heater. Steam at high pressure enters a finned condensing coil, the condensate flowing into a trap from which it is admitted to the vapor coil entering heater. Here it flashes into steam at low pressure and enters the vapor condensing coil for final condensation. Advantages of this system, developed by Niagara Blower Company, N. Y., are that the apparatus is simple and economical to install because it requires no pressure reducing stations; live steam is not wasted as in dump-trap or hot-well methods; efficiency is high with maximum utilization of all latent and sensible heat; condensate is returned in even flow at low pressure temperature and the vacuum return is vapor-free.

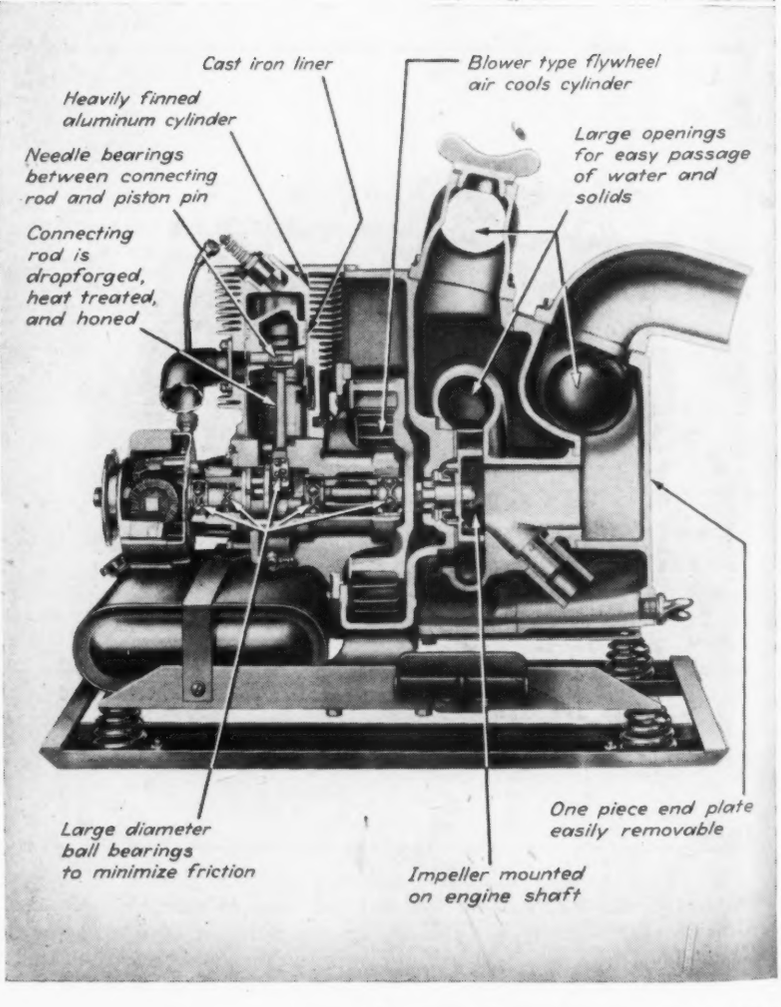


Top leg of vapor condensing coil is finned and steeply pitched; lower leg is relatively level and leads to the condensate return header. In this header is a weir that keeps liquid level high in the lower leg, where the liquid is in contact with the coldest portion of the air steam. Condensate is therefore subcooled below its relative return temperature. The weir has a bleed hole that drains the coil upon shutdown. Coils are of copper with aluminum fins.

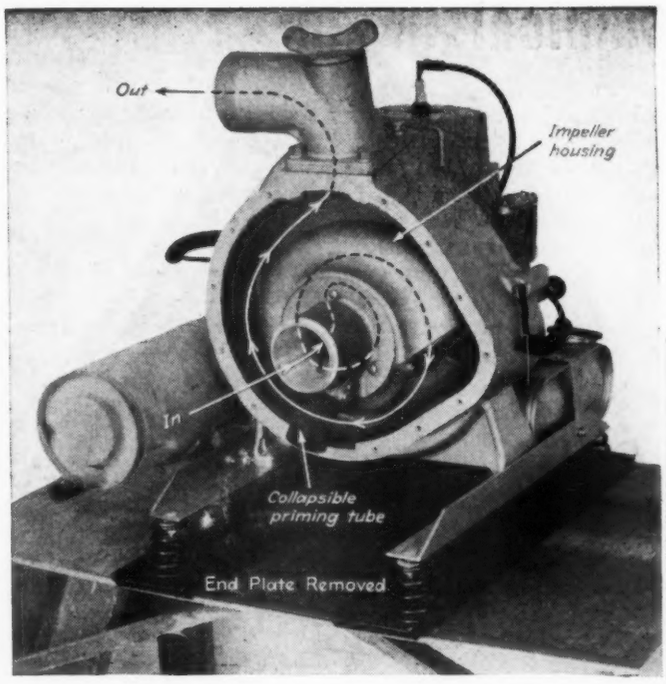
Compact Self-Priming Pump



AUTOMATIC CONTROL of self-priming and simple, maintenance-free construction are features of the 95 lb. centrifugal pump built by Homelite Corporation, Port Chester, N. Y. A compact, light unit is achieved by building the pump around the engine shaft and using aluminum alloy for all large castings. Capacity is 15,000 g.p.h. with seepage as low as 1 g.p.h. Engine is a single cylinder, air-cooled, two-cycle gasoline type. High tension, flywheel type magneto is dust and waterproof. Cylinder and piston are aluminum with a cast iron liner in the cylinder. The one valve is a self grinding, independent rotary disk type. A mechanical governor is built-in. Fuel consumption is 1¾ g.p.h. at full load.

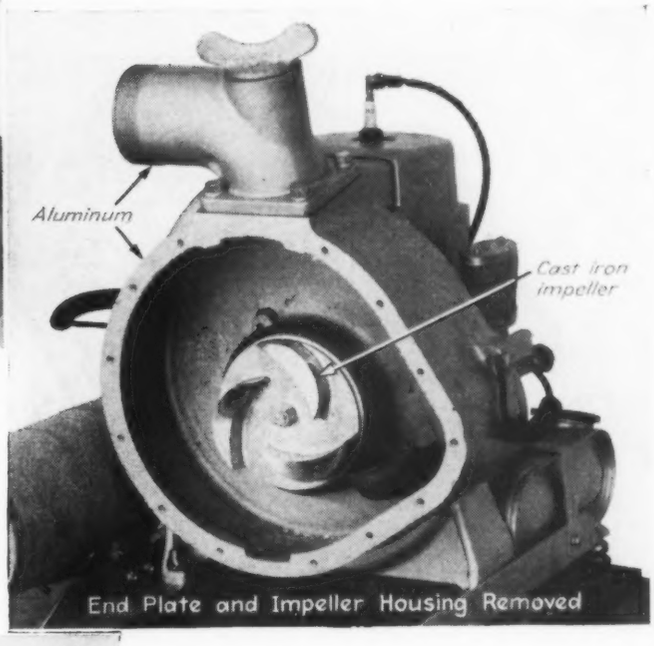

Cast iron liner
Blower type flywheel air cools cylinder
Heavily finned aluminum cylinder
Large openings for easy passage of water and solids
Needle bearings between connecting rod and piston pin
Connecting rod is dropforged, heat treated, and honed
One piece end plate easily removable
Large diameter ball bearings to minimize friction
Impeller mounted on engine shaft

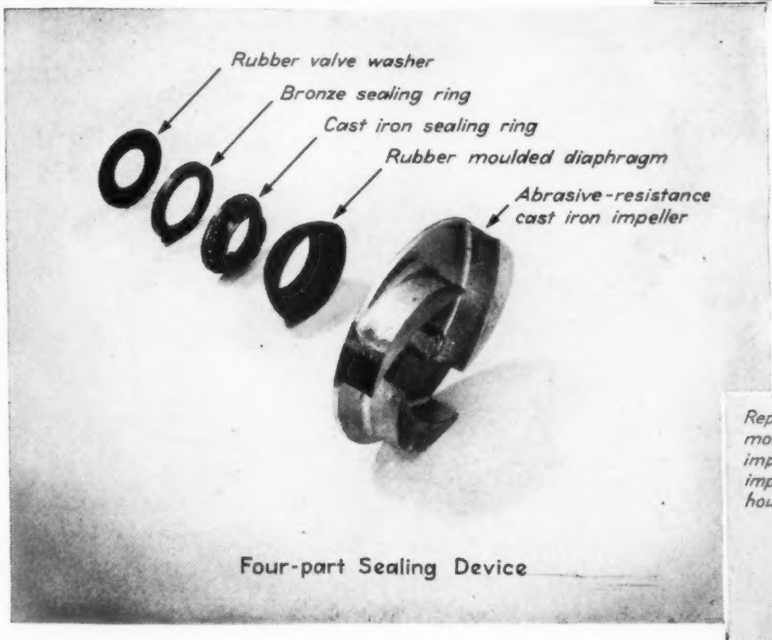
One-piece casting serves as pump body and engine mounting. Only moving part in the pump is the impeller mounted on the end of the engine shaft. Shaft seal is mounted directly into the body. The seal comprises four easily-replaceable parts. Heat resisting recess above ring grooves on the aluminum alloy piston reduces carbon formation and ring sticking. The ignition is shielded from rain and splashing.



Fast self-priming is achieved through a simple collapsible priming tube of rubber. The tube automatically opens for priming or seepage handling and closes for volume pumping. It provides a 25 ft. suction lift in 100 sec.; a 5 ft. suction lift in 15 sec. Total head is 80 ft., including friction. Discharge and suction are each through three inch diameter orifices.

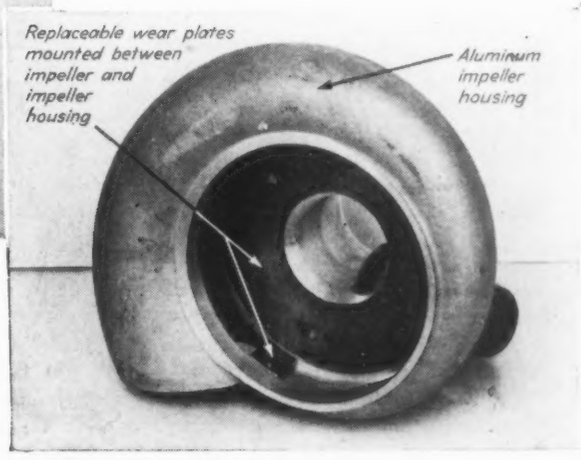
Open vane impeller design and large clearances enable the pump to handle heavy sediments. The flow of water directed across bottom of the pump body prevents an accumulation of sediment or solids. These materials are carried out with the discharge. The impeller was manganese bronze but tests with abrasive sand-water mixtures show that the new cast iron impeller has four times the life of bronze types.





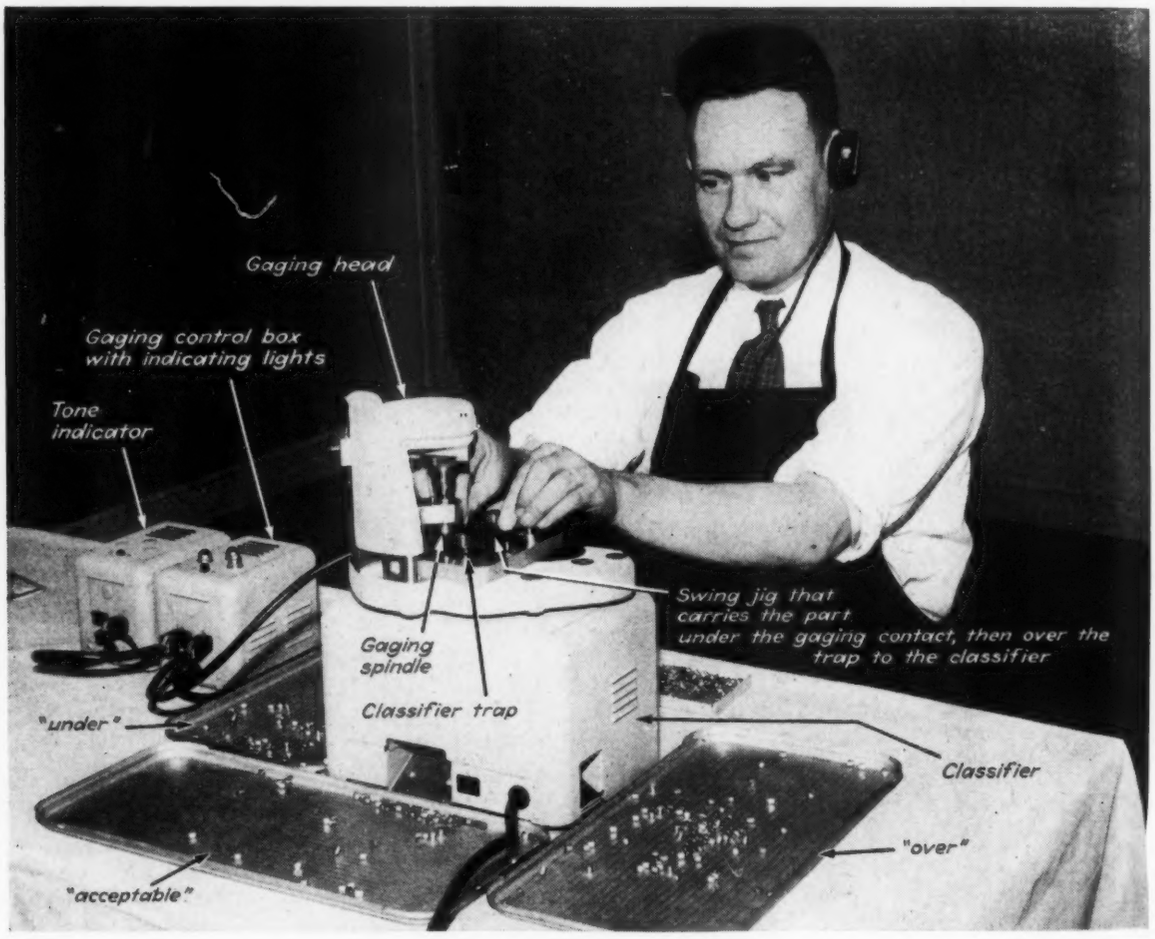
Four-part sealing device is effective and easily replaceable by components. Sealing rings are self adjusting. No service lubrication is required. There are no bearings on the pump end, therefore avoiding that source of potential trouble.

To reduce wear and maintain the pumping action, replaceable wear plates of a special abrasive-resistant cast iron are mounted between impeller and impeller housing. An aluminum alloy, specially designed for abrasion resistance, is cast to form the housing. The one piece impeller housing is replaceable as a unit.

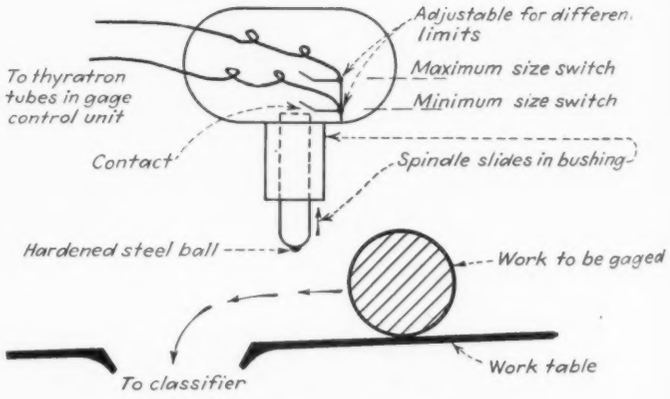


Electro-Mechanical Gaging

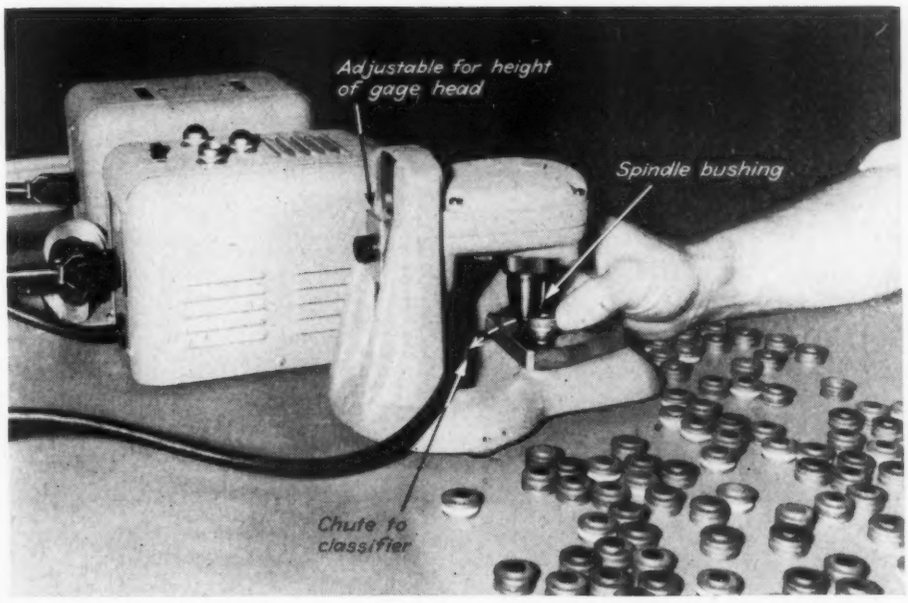
ELECTRONIC SORTING GAGES frequently are affected by adjacent magnetic fields, variations in line voltage and similar electrical disturbances. Mechanical gaging suffers from inaccuracies caused by wear in sliding and rotating parts. The Limitron, produced by Arma Company, Brooklyn, combines mechanical gaging with electrical measuring and sorting to overcome the weaknesses in either system. The Limitron is a comparator type, universal sorting gage that classifies a dimension as "over," "under" or "acceptable". It is designed for production gaging of dimensions up to 1½ in. to limits within 0.0001 to 0.0300 in. and at rates up to 7,000 pieces per hour. The small gaging head can be mounted in any plane and various automatic feeding devices can be applied. In above picture, a blind man is sorting parts. The earphone brings him one of three audible signals to indicate the classification of each part as it is sorted. Ordinarily red, green and yellow lights flash on the control unit for oversize, acceptable, and undersized parts.



The Limitron comprises a gaging head, gaging control and classifier. Gaging head carries a tool steel pin that slides in a hardened bushing. Contact with the work is made by a small chrome steel ball fixed on the end of the spindle. Gaging pressure is adjustable between 4 oz. and 1 lb. As the piece is pushed under the spindle, it forces the spindle into the head. Depending on amount of displacement, the spindle will fall short of reaching the minimum size switch or will close one or two limit switches that previously have been set with a master, to indicate minimum and maximum limits. If spindle displacement closes a switch, a circuit is completed to a thyratron tube. The power amplifier then operates the corresponding torque motor in the classifier unit to operate a gate.

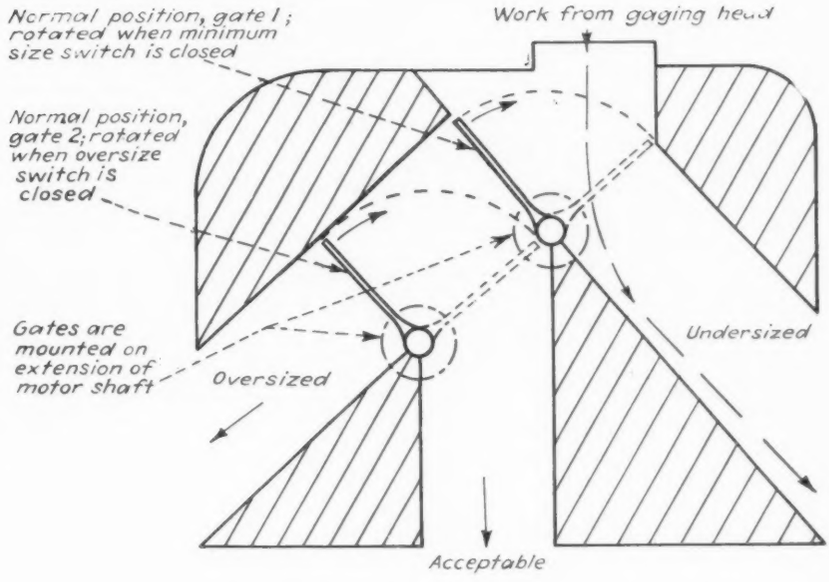


Schematic Sketch of Gaging Head

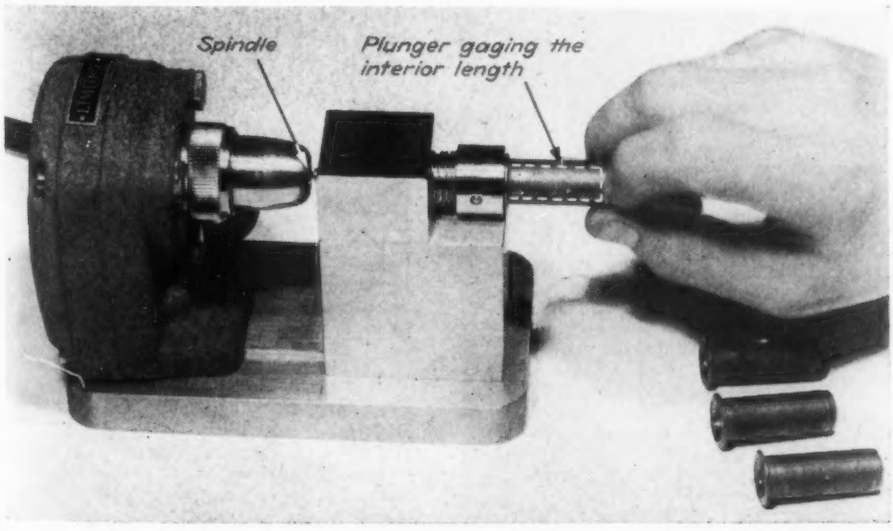


Adjustments are made inside the gage head, free from possibility of tampering. Once the switches are set to master pieces, the unit can be operated by anyone. Possibilities for human error in reading the gage each time or placing the piece in the right sorting box are avoided. Chutes and gates are lined with a nylon-based resin bonded material so that the parts do not touch metal as they fall through the classifier.

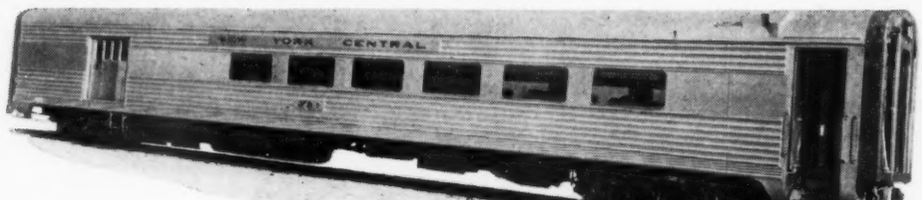
A gate, or flipper, is mounted on each torque motor shaft. Sketch shows how chutes are opened by action of the motors. As a safety device, chutes are arranged so that power failure or operator negligence will leave only the minimum-size reject path open. Solenoids were not used to actuate gates because the linkage required to translate straight-line into rotary motion introduces a source of wear. Also, solenoid force increases as the solenoid travels into final position, whereas a torque motor has the opposite force-travel characteristic and eases the gate closed, rather than slams it into position.


Normal position, gate 1; rotated when minimum size switch is closed
Work from gaging head
Normal position, gate 2; rotated when oversize switch is closed
Gates are mounted on extension of motor shaft
Oversized
Undersized
Acceptable

Schematic Sketch of Classifier Unit



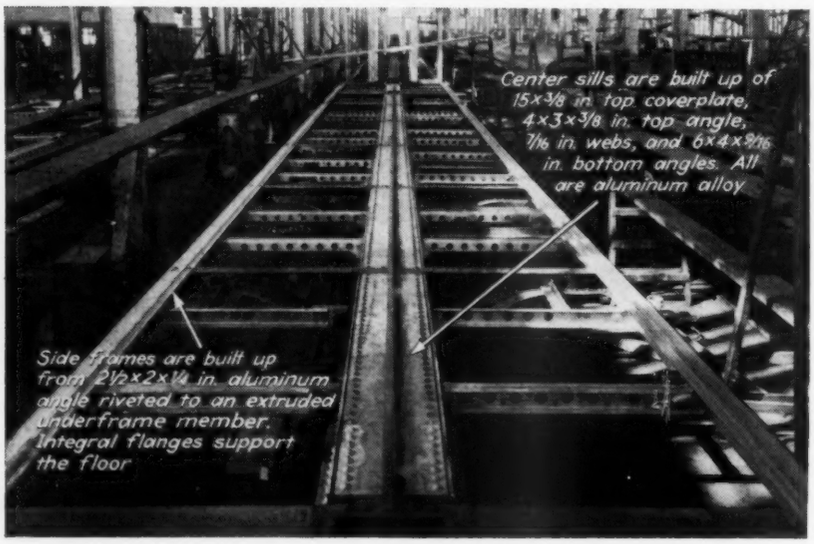
Internal dimensions are gaged on a plunger arranged as shown above. The Limitron can be set up to gage in many different ways, and in combination with other units. A counting unit that records acceptable and oversize pieces is available, as well as a complete worktable and turntable feed unit for automatic gaging.



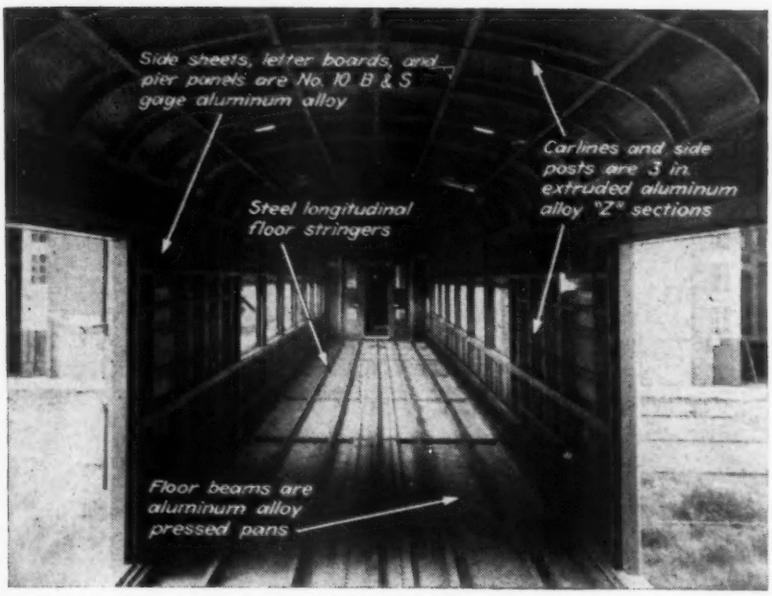
Coach-Baggage Car

Constructed of Aluminum Alloys

WEIGHT SAVINGS of approximately 34 percent as compared to conventional steel construction are realized in these new aluminum coach-baggage cars. The 24 seats are full-reclining and rotating, with individual foot rests. Interior finish has no exposed screw heads or battens. Ceiling is a new, full floating type, housing a central air diffuser for the air conditioning system. Individual lights are recessed in bottom of luggage rack supports. For ease of cleaning, rust colored linoleum covers bulkheads, locker fronts, floors and wainscoting. Baggage car has a unit heater of steam coil and blower fan type. Car exterior has anodized aluminum fluted extrusions and snap-on mouldings above and below the wide view windows. Cars were designed to meet both U.S. Post Office and A.A.R. passenger equipment specifications. Built by American Car & Foundry Company, St. Charles, Mo. with interior styling by Henry Dreyfuss.

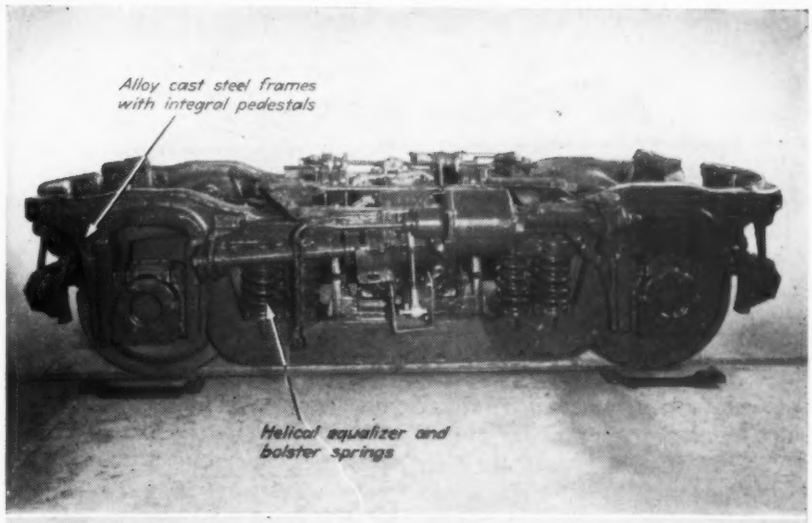

Center sills are built up of 15×3/8 in. top coverplate, 4×3×3/8 in. top angle, 7/16 in. webs, and 6×4×9/16 in. bottom angles. All are aluminum alloy
Side frames are built up from 2½×2×¼ in. aluminum angle riveted to an extruded underframe member. Integral flanges support the floor

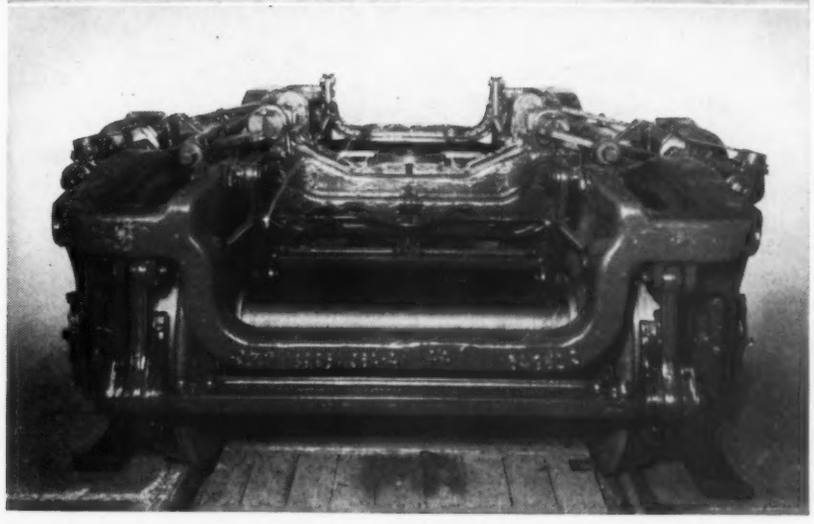
Car body framing is aluminum except for bolsters, cross bearers and end sills. Cast steel buffer extends through the bolster and is secured to center sill by riveting. This joint is protected against corrosion by hot zinc metallizing. Bolsters and cross bearers are built up of welded steel. Special extruded aluminum alloy sections are used for many framing members. Rivets not covered by fluting have an anodized, low oval head of special design. End frame has two 8 in. aluminum alloy extruded I-beam sections with web thickness in lower portion to meet A.A.R. shear resistance requirements. These sections extend into pockets in buffer casting, likewise supported at top by anti-telescoping construction of a 10 in. aluminum channel at stub end and truss girder construction at the vestibule end.


Side sheets, letter boards, and pier panels are No. 10 B & S gage aluminum alloy
Carlines and side posts are 3 in. extruded aluminum alloy "Z" sections
Steel longitudinal floor stringers
Floor beams are aluminum alloy pressed pans

Floor sheets, 5/8 **in.** thick, are fastened to longitudinal stringers and 1/8 in. composition flooring is applied on top of sheets. No. 16 B. & S. gage galvanized false floor is secured to bottom of stringers, to side sill extrusions and center sill top cover. Inside of all outside sheets and outside of inside sheets are sprayed with insulating compound. Wires hold the 3 in. of insulation in place on floor, sides, ends and roof of car. Tape breaks the contact between framing and inside finish.

Trap doors are aluminum with stainless steel, super-diamond floor plates, operating with four folding steps of aluminum and stainless steel to form a continuous part of the skirting when in closed position. A short, fluted skirt flares down at ends of car to conceal step sides. It is held by hidden bolts that permit removal and replacement entirely from the exterior of the car without disturbing any other parts of the car body.





Trucks are single drop equalizer type with 5½x10 in. journals at passenger end and 6x11 in. journals at baggage end. The 36 in. rolled steel wheels are ground after mounting to insure concentric treads. Sound-deadening materials isolate noises from car at bolster springs, truck center plates, body side bearings and at moving parts of coupler and buffer mechanism.

Internal Floating Friction Shoes

Analysis and Application

A. C. RASMUSSEN
Consulting Engineer

Formulas for computing the resultant force on floating shoes, the resultant normal pressure and the point of maximum intensity of lining pressure. An example illustrates the application of the formulas. Two tables and a chart are included as aids to simplifying and reducing the numerical work of design calculations.

INTERNAL floating friction shoes designate a class of brake or clutch shoes that are not held definitely in position by a rigid hinge pin mounted on a carrier or driving plate. Instead, a link serves as the means for connecting the floating shoe to the driving plate. Likewise, the actuating force is applied to the shoe through a link. Fig. 1 shows a typical floating shoe with the supporting and actuating links.

Although the shoe is pressed against the inner face of the drum by two widely separated forces, T and P, the effect is the same as that produced by a single force; namely, the resultant of three forces: The actuating force, T, the reaction of the driving pin P, and ηR, the frictional force between the shoe and the drum. Before proceeding with the analysis of the forces caused by applying an actuating force to a shoe, the distribution of the pressure applied to a shoe will be determined. Let R equal the resultant force with which the shoe presses against the drum. For equilibrium this force must pass through the center of pressure of the shoe against the drum. Assume for the present that this force is applied at the center of the shoe, the analysis when the load is applied off-center being treated later.

The intensity of pressure at the center of the shoe is to be determined because it must be kept less than the maximum permissible lining pressure.

ASSUMPTION. Let q_1 be the intensity of normal pressure at the center of the shoe. Then the intensity of pressure at any other point q_x at an angular distance α, Fig. 2, from the center of contact of the shoe is

$$q_x = q_1 \cos \alpha \qquad (1)$$

The pressure on the small area of r $\Delta\alpha$ length and b width is,

$$q_x br\, \Delta\alpha = q_1 br \cos \alpha\, \Delta\, \alpha \qquad (2)$$

The component of this increment of pressure parallel to R is

$$\Delta R = q_1\, br \cos^2\alpha\, \Delta\alpha \qquad (3)$$

Integrating Equation (3) between the limits $-\frac{1}{2}\theta$ and $\frac{1}{2}\theta$ gives

$$R = q_1 br\, (\theta + \sin \theta)/2 \qquad (4)$$

from which

$$q_1 = \frac{2\,R}{br\,(\theta + \sin\theta)} \qquad (5)$$

This maximum unit pressure, q_1, at the center of the shoe is a normal pressure. Unit normal pressure at any other point from Equation (1)

$$q_x = \frac{2\,R\cos\alpha}{br\,(\theta + \sin\theta)} \qquad (6)$$

Let C be the total normal pressure. Then the normal pressure on the small area, $br\Delta\alpha$ is,

$$\Delta C = q_x br\, \Delta\alpha = \frac{2\,R\cos\alpha\,\Delta\alpha}{\theta + \sin\theta} \qquad (7)$$

Integrating Equation (7) between the limits $-\frac{1}{2}\theta$ and $\frac{1}{2}\theta$ gives

$$C = \frac{4\,R\sin\frac{1}{2}\theta}{\theta + \sin\theta} \qquad (8)$$

The relation of C to R can be denoted as c, or

$$c = C/R = \frac{4\sin\frac{1}{2}\theta}{\theta + \sin\theta} \qquad (9)$$

Values of c for θ from 5 to 180 degrees are given in Table I in 5-deg. increments.

Since the normal pressure between the friction surfaces is the cause of resistance to movement through friction, the resistance to movement is greater with a greater pressure. It is apparent from Equation (9), which numerically is greater than unity for all values of θ between zero and 180 deg., that the total normal pressure is greater than R. Therefore the product of the coefficient of friction μ and R is not the total frictional force between the friction surfaces, but rather the product μC is the total frictional force. From Equation (9) μC equals μRc so that μc can be taken as the effective coefficient of friction with respect to R. If η equals μc,

Table I—Numerical Values of c Ratios for θ from 5 to 180 degrees

θ deg.	c	θ deg.	c	θ deg.	c	θ deg.	c
5	1.0003	50	1.0316	95	1.1111	140	1.2179
10	1.0013	55	1.0382	100	1.1224	145	1.2289
15	1.0029	60	1.0454	105	1.1340	150	1.2392
20	1.0051	65	1.0531	110	1.1458	155	1.2485
25	1.0079	70	1.0615	115	1.1579	160	1.2567
30	1.0114	75	1.0704	120	1.1701	165	1.2635
35	1.0155	80	1.0798	125	1.1824	170	1.2688
40	1.0203	85	1.0898	130	1.1945	175	1.2721
45	1.0256	90	1.1002	135	1.2064	180	1.2732

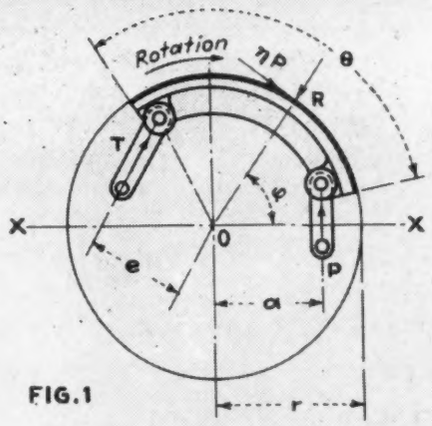


FIG.1

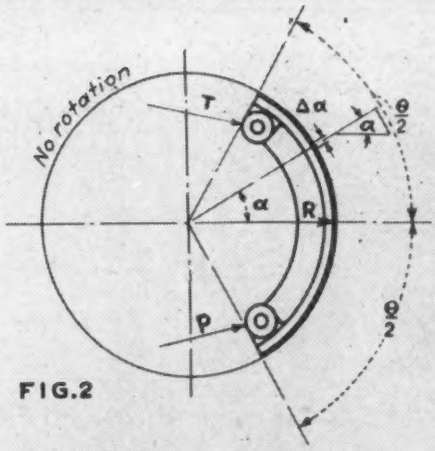


FIG.2

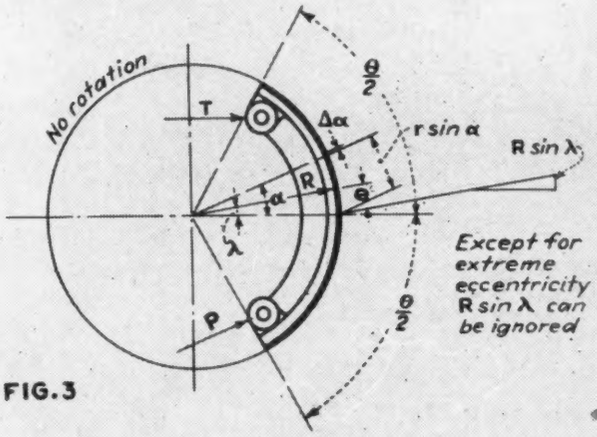


FIG.3

Fig. 1—Floating shoe in which T is the link through which the actuating force is applied and P is the supporting link.

Fig. 2—System of forces acting on shoe in such a manner that the resultant force R passes through central point of shoe.

Fig. 3—Usually the resultant force R passes through some off-center position.

ηR represents the frictional force on internal floating friction shoes. The value of c depends entirely upon the extent of the angle of contact between the friction surfaces. As the length of contact of a shoe is known or can be approximated, a reasonably close value of η can be decided upon before making the complete calculations for a shoe. Should the shoe work out to have a different length, corrections in the calculations can then be made.

Since the resultant pressure against the shoe seldom intersects the friction surface at the center of the shoe, the off-center position of R, Fig. 3, will represent these force relations. The point of greatest unit pressure is also off-center and is at some point beyond the point at which R intersects the friction surface.

ASSUMPTION. It is assumed that the intensity of the normal pressure is proportional to the distance that the normal is from the central point of contact of the shoe against the drum.

Let p_1 be the normal unit pressure at the tip of the shoe resulting from the moment of the off-center force. Then unit pressure at unit distance from the center of contact is $p_1/r \sin \frac{1}{2}\theta$. The unit pressure at any point x is,

$$p_x = p_1 \sin \alpha / \sin \tfrac{1}{2}\theta \qquad (10)$$

The pressure on a small area of $r\,\Delta\alpha$ length and b width is,

$$\Delta P = br\,\Delta\alpha\, p_1 \sin \alpha / \sin \tfrac{1}{2}\theta \qquad (11)$$

The moment of this increment pressure about the middle point of the shoe is:

$$\Delta M = r \sin \alpha\, \Delta P \qquad (12)$$

Combining Equations (11) and (12) to eliminate ΔP and integrating the variable angle α between the limits $-\frac{1}{2}\theta$ and $\frac{1}{2}\theta$ yields

$$M = p_1 br^2 (\theta - \sin\theta)/2 \sin \tfrac{1}{2}\theta \qquad (13)$$

If e is the eccentricity of the force R with respect to the center of shoe,

$$M = Re \qquad (14)$$

But with respect to the center of the shoe as the origin of moments, the force R causes the intensity of pressure to increase on one side and decrease on the other. Combining Equations (13) and (14) and solving for p_1 gives

$$p_1 = \pm \frac{2\,Re \sin \frac{1}{2}\theta}{br^2 (\theta - \sin\theta)} \qquad (15)$$

Substituting the p_1 of Equation (15) into Equation (10), there is obtained

$$p_x = \pm \frac{2\,Re \sin \alpha}{br^2 (\theta - \sin\theta)} \qquad (16)$$

The stress, q_x, at any point resulting from the direct load is given by Equation (6) while the additional stress, p_x, caused by the off-center loading is given by Equation (16). The combined stress S_x, therefore, is

$$S_x = q_x \pm p_x = \frac{2R}{br^2}\left(\frac{r \cos \alpha}{\theta + \sin\theta} \pm \frac{e \sin \alpha}{\theta - \sin\theta}\right) \qquad (17)$$

If λ is the eccentricity angle of force R with respect to center of shoe, Fig. 3, e equals $r\sin\lambda$ so that Equation (17) can be written

$$S_x = \frac{2R}{br}\left(\frac{\cos \alpha}{\theta + \sin\theta} \pm \frac{\sin \lambda \sin \alpha}{\theta - \sin\theta}\right) \qquad (18)$$

When the eccentricity of R is excessive, that tip of the shoe the farther from the center of load application will raise away from the drum surface to become ineffective. While it is evident that R should be at the center of the shoe for most effective results, this condition cannot always be obtained easily. But the friction surface can be moved ahead or back-

Table II—Numerical Values of Functions of θ at 5 Degree Increments

θ deg.	θ radians	$\theta + \sin\theta$	$\theta - \sin\theta$	$\frac{\theta+\sin\theta}{\theta-\sin\theta}$	$\frac{\theta-\sin\theta}{\theta+\sin\theta}$
5	0.0873	0.0872	0.0001107	1575.3450	0.0006348
10	0.1745	0.3482	0.0008848	393.5361	0.0025411
15	0.2618	0.5206	0.0029803	174.6842	0.0057246
20	0.3491	0.6911	0.0070457	98.0861	0.010195
25	0.4363	0.8590	0.013714	62.6329	0.015966
30	0.5236	1.0236	0.023599	43.3751	0.023055
35	0.6109	1.1844	0.03729	31,7640	0.031482
40	0.6981	1.3409	0.05534	24.2288	0.041273
45	0.7854	1.4925	0.07829	19.0635	0.052456
50	0.8727	1.6387	0.1066	15.3696	0.06506
55	0.9599	1.7791	0.1408	12.6374	0.07913
60	1.0472	1.9132	0.1812	10.5602	0.09469
65	1.1345	2.0408	0.2282	8.9446	0.11180
70	1.2217	2.1614	0.2820	7.6636	0.13049
75	1.3090	2.2749	0.3431	6.6311	0.15081
80	1.3963	2.3811	0.4115	5.7869	0.17280
85	1.4835	2.4797	0.4873	5.0883	0.19653
90	1.5708	2.5708	0.5708	4.5039	0.22203
95	1.6581	2.6543	0.6619	4.0103	0.24936
100	1.7453	2,7301	0.7605	3.5898	0.27857
105	1.8326	2.7985	0.8667	3.2291	0.30969
110	1.9199	2.8596	0.9802	2.9174	0.34277
115	2.0071	2.9134	1.1008	2.6466	0.37784
120	2.0944	2.9604	1.2284	2.4100	0.41493
125	2.1817	3.0008	1.3625	2.2024	0.45405
130	2.2689	3.0350	1.5029	2.0194	0.49519
135	2.3562	3.0633	1.6491	1.8576	0.53834
140	2.4435	3.0862	1.8007	1.7139	0.58345
145	2.5307	3.1043	1.9572	1.5861	0.63046
150	2.6180	3.1180	2.1180	1.4721	0.67928
155	2.7053	3.1279	2.2826	1.3703	0.72977
160	2.7925	3.1345	2.4505	1.2791	0.78177
165	2.8798	3.1386	2.6210	1.1975	0.83507
170	2.9671	3.1407	2.7934	1.1243	0.88942
175	3.0543	3.1415	2.9672	1.0587	0.94451
180	3.1416	3.1416	3.1416	1.0000	1.00000

ward with respect to the link pin positions so that the eccentricity is minimized. Certainly, the amount of eccentricity should not be more than that with which the unit pressure is reduced to zero at the farther tip of the shoe. The limit of eccentricity can be found by equating Equation (18) to zero, the sign of the second term within the parentheses being negative and the angle α being $\frac{1}{2}\theta$. If this angle is denoted by λ_1, there is obtained

$$\sin\lambda_1 = \frac{1}{\tan\frac{1}{2}\theta}\,\frac{\theta-\sin\theta}{\theta+\sin\theta} \qquad (19)$$

With R eccentric to the center of the shoe, the maximum pressure will be at some point slightly beyond the point where R intersects the friction surface. Referring to Fig. 4 let ψ be the angle between the center of the shoe and the point of maximum pressure. The value of ψ is obtained by differentiating Equation (18) with respect to α, equating the differential to zero and solving for the value of α. This value of α is designated as ψ. Thus

$$\tan\psi = \frac{\sin\alpha}{\cos\alpha} = \sin\lambda\,\frac{\theta+\sin\theta}{\theta-\sin\theta} \qquad (20)$$

See Table II for values of the θ functions of Equation (20) corresponding to values of θ from 5 to 180 deg. in 5-deg. increments.

Proceeding with the analysis of the floating friction shoe, the following nomenclature is used:

- T = actuating force, the line of action of which passes through the centers of the two pins of the actuating link;
- P = reaction of supporting pin acting along line passing through centers of the supporting link pins;
- R = reaction of drum against pressure of shoe;
- μ = coefficient of friction of lining;
- η = equivalent coefficient of friction of lining when used on a shoe segment;
- β = inclination of T with respect to axis XOX, Fig. 5;
- a = radius of force P;
- e = radius of force T;
- θ = angular length of contact between shoe and drum;
- λ = eccentricity of R with respect to center of contact.

Note that in Fig. 5 and in the following analysis, the base line XOX is taken prependicular to P. The force P therefore has no horizontal component. This is done to keep the equations as simple as possible.

By inspection of Fig. 5 the following equation can be written for the vertical forces:

$$P + T\sin\beta - R\sin\phi + \eta R\cos\phi = 0 \qquad (21)$$

Let

$$k_\phi = \sin\phi + \eta\cos\phi \qquad (22)$$

then

$$P + T\sin\beta - Rk_\phi = 0 \qquad (23)$$

Similarly the following equation can be written with respect to the horizontal components of the forces. Let

$$T\cos\beta - R\cos\phi + \eta R\sin\phi = 0 \qquad (24)$$

$$h_\phi = \cos\phi - \eta\sin\phi \qquad (25)$$

Then

$$T\cos\beta - Rh_\phi = 0 \qquad (26)$$

The equation for the moment about the center of rotation can be written

$$Pa = Te + \eta Rr \qquad (27)$$

Multiplying all terms of Equation (23) by a and substituting therein the value of Pa from Equation (27) gives

$$Te + \eta Rr + Ta\sin\beta - Rak_\phi = 0 \qquad (28)$$

Combining Equations (25) and (28) to eliminate R and T, gives upon rearrangement of terms

$$k_\phi - h_\phi\,\frac{(e + a\sin\beta)}{a\cos\beta} = \frac{\eta r}{a} \qquad (29)$$

Let

$$m = (e/a)\sec\beta + \tan\beta \qquad (30)$$

and

$$s = \eta r/a \qquad (31)$$

so that Equation (29) can be written simply as

$$k_\phi - h_\phi m = s \qquad (32)$$

If k_ϕ and h_ϕ in Equation (32) are replaced by the equivalent functions of ϕ as defined by Equations (22) and (25), respectively, there is obtained

$$(1 + m\eta)\sin\phi - (m - \eta)\cos\phi = s \qquad (33)$$

which expresses ϕ in terms of known quantities. The $\cos\phi$ can be eliminated and the resulting quadratic equation in $\sin\phi$ solved for the angle. The corresponding quantities for k_ϕ, h_ϕ, R and P can be calculated from Equations (22), (25), (26) and (27), respectively.

As a check, Equation (23) can be

used. Note should be made that when the inclination of T is downward, the angle β is negative and that the proper choice of sign needs to be used wherever functions of β appears in the calculations.

In working out solutions, a standard procedure has been developed. It is illustrated by the example at the end. With such a procedure, the solution of design problems becomes methodical and it is unnecessary to reason out each step, thereby saving time. A further advantage is obtained by working out the solutions in terms of one pound actuating force and unit radius, giving quantity coefficients. Comparisons of results can be made more readily by comparing coefficients. Final results are then obtained by multiplying the coefficients by T in pounds and r in inches.

The floating type of shoe is used principally because of the ease with which it can be mounted. At the same time the wear can be made to occur at the center of the shoe rather than at one end or the other. Self-energization can result only when motion between the friction surfaces occurs or when there is a tendency for motion to occur. Therefore the ratio of self-energization can be determined by comparing the value of R obtained when motion occurs with a value of R, designated as R_o, that results from the same actuating force but with no motion. Another way of obtaining the same result is to compare the value of R obtained with motion to the value of R_o obtained with motion but with zero friction. With zero friction, it is evident that P will be less in value and therefore R_o and its inclination will be different. It is necessary therefore to calculate R_o and the corresponding value of ϕ, which will be called ϕ_o. The equations, similar to Equations (21), (24) and (27), are

$$P_0 + T\sin\beta - R_0\sin\phi_0 = 0 \qquad (34)$$

$$T\cos\beta - R_0\cos\phi_0 = 0 \qquad (35)$$

and

$$P_0 a = Te \qquad (36)$$

These three equations when solved for ϕ_o give

$$\tan\phi_0 = (e/a)\sec\beta + \tan\beta \qquad (37)$$

Note that value of $\tan\phi_o$ is the value of m as given by Equation (30).

Upon substituting this value of ϕ_o into Equation (35) the value of R_o becomes known, that is

$$R_0 = T\cos\beta/\cos\phi_0 \qquad (38)$$

From Equation (26)

$$R = T\cos\phi/h_\phi \qquad (39)$$

and consequently the self-energization ratio E is given by

$$E = R/R_0 = \cos\phi_0/h_\phi \qquad (40)$$

The ratio of self-energization can also be determined by comparing the moment of T_o required to produce a given value of R_o with no motion to the moment of T required to produce a value of R equal to R_o but with motion.

Taking moments about point (1), Fig. 6,

$$T_0(e + a\sin\beta) = R_0 a\sin\phi_0 \qquad (41)$$

But from Equation (28)

$$T(e + a\sin\beta) = R(ak_\phi - \eta r) \qquad (42)$$

so that

$$E = T_0/T = R_0 a\sin\phi_0/R(ak_\phi - \eta r) \qquad (43)$$

Since R_o is to be taken equal to R,

$$E = \sin\phi_0/(k_\phi - \eta r/a) \qquad (44)$$

When $\eta r/a$ equals k_ϕ, the denomina-

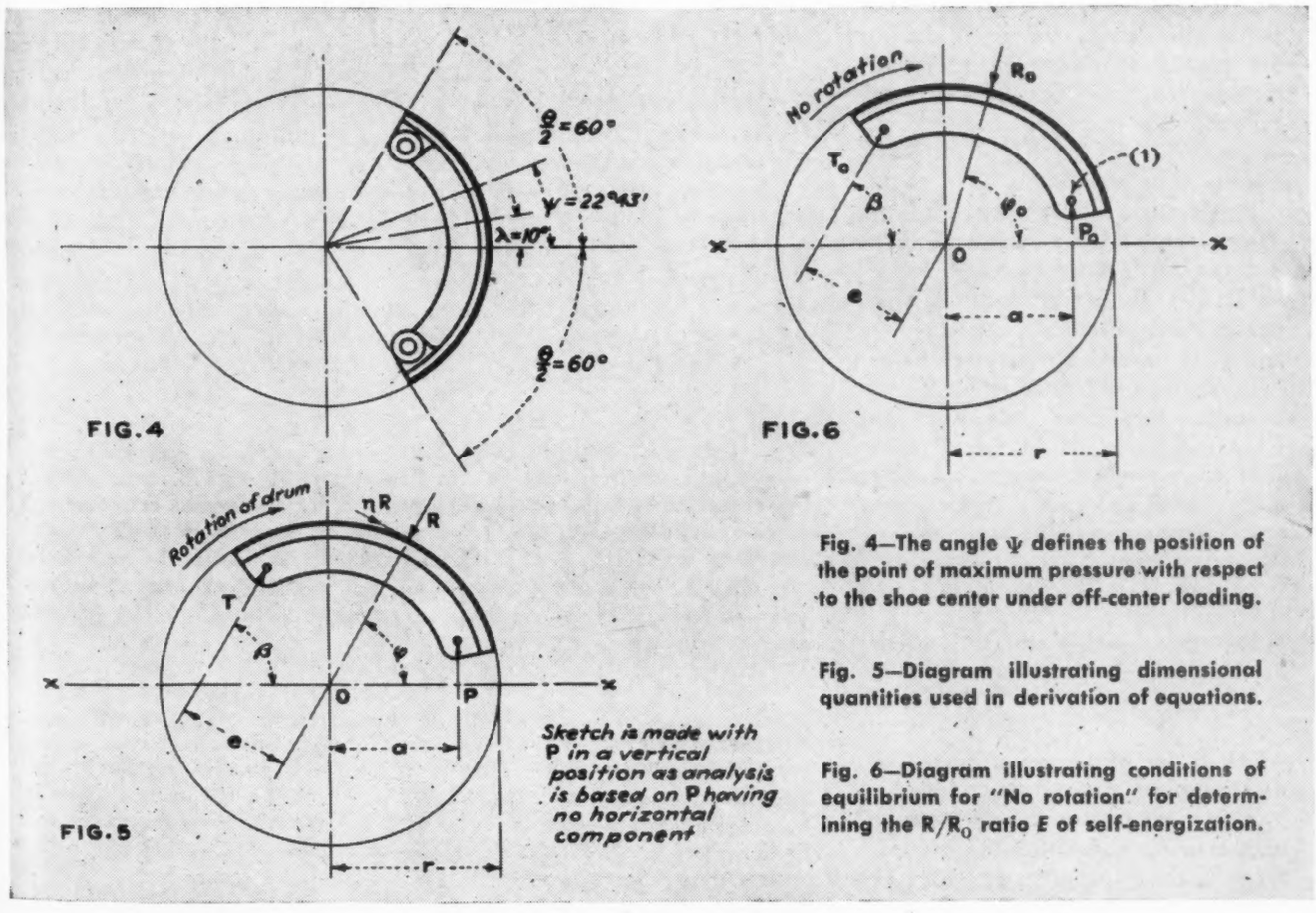


Fig. 4—The angle ψ defines the position of the point of maximum pressure with respect to the shoe center under off-center loading.

Fig. 5—Diagram illustrating dimensional quantities used in derivation of equations.

Fig. 6—Diagram illustrating conditions of equilibrium for "No rotation" for determining the R/R_0 ratio E of self-energization.

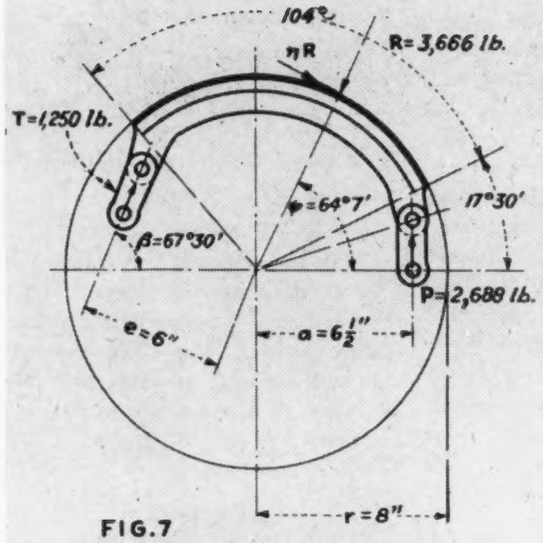


FIG.7

Fig. 7—Given dimensions and some computed values of the numerical example.

Fig. 8—Chart of c values showing that these values are greater than unity for all values of θ between zero and 180 degrees.

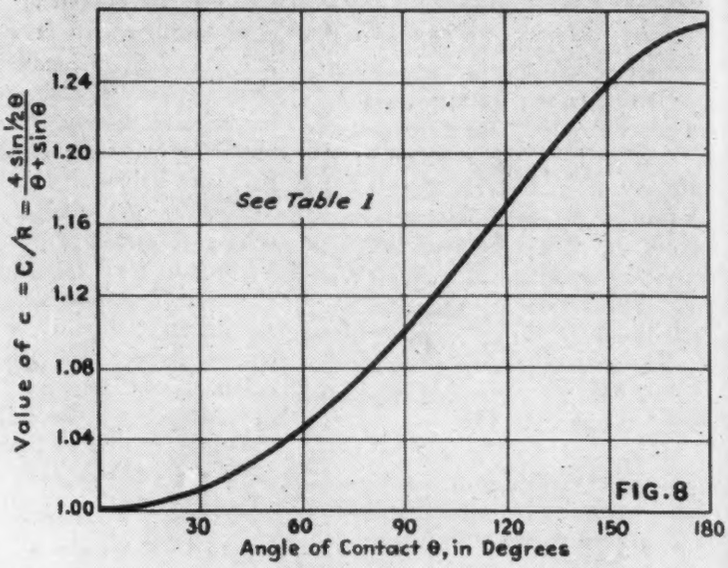


FIG.8

tor reduces to zero and the condition for locking occurs.

EXAMPLE. Determine the braking torque and the maximum unit lining pressure of the shoe shown in Fig. 7. Also determine E, the self-energization ratio. The drum is 16 in. in dia. and 3 in. wide. The radius of the supporting link is 6.5 in. and the radius of the actuating link is 6 in. A coefficient of friction of 0.30 is assumed. The angle β is 67 deg. 30 min., the angle θ is 104 deg. An actuating force of 1,250 lb. is applied.

From Table I for θ of 104 deg. c is 1.1316. By definition η equals μc equals 0.34. From Equations (30) and (31) m and s are found to be 4.8264 and 0.4185, respectively. These quantities substituted into Equation (33) make ϕ equal 64 deg. 7 min. The corresponding values of k_ϕ and h_ϕ are 1.0481 and 0.1305, respectively.

From Equation (39)

$$R = T \cos \beta / h_\phi$$
$$= 1{,}250 \times 0.3827/0.1305 = 3{,}666 \text{ lb.}$$

Now the braking torque is given by the product

$$\eta R r = 0.34 \times 3{,}666 \times 8 = 9{,}971 \text{ in.-lb.}$$

which is one of the quantities to be determined.

To get the maximum unit pressure on the lining Equation (18) is used. Some of the θ quantities of Table II are substituted into Equation (18) directly; others are introduced indirectly through Equations (19) and (20). Thus

$$S_x = \frac{2R}{br^2}\left(\frac{r \cos \alpha}{\theta + \sin \theta} + \frac{\sin \lambda \sin \alpha}{\theta - \sin \theta}\right)$$
$$= \frac{7{,}332}{24}\left(\frac{0.9548}{2.7854} + \frac{0.0944 \times 0.2971}{0.8448}\right)$$
$$= 115 \text{ lb. per sq. in.}$$

Had Equation (5) been used, that is the on-center condition of loading, the calculated intensity of pressure would have been 110 lb. per sq. in. instead of 115 lb. per sq. in. For this example therefore the more precise analysis was hardly worth the extra effort. On the contrary, should the eccentricity be appreciable and the friction surface cannot be moved relative to the hinge pins to a more on-center position, then the maximum unit pressure should be calculated. If the unit pressure is excessive for the lining being used, the actuating force must be reduced with the result that the braking torque is reduced. On the other hand, a harder lining can be used but harder linings usually have a lower coefficient of friction and again a lower braking torque is obtained. Consequently every effort should be made to move the center of the friction surface as near as possible to the point where the resultant R intersects the friction surface.

* * * *

ONE of the greatest troubles with early gas engines was in finding a suitable way to ignite the fuel mixture. An Englishman, William Barnett, invented in 1838, what he called an "ignition cock." This igniter was in reality two flames; one inside and the other outside the cylinder. As the explosion put out the inside flame a small port opened. Through this port the inside flame was relighted by the one outside. In the year 1857, this complicated mechanism was replaced by the electric igniter, invented by the Italians, Barsanti and Matteucci.

* * * *

ENTERPRISING BURGLARS overcame the obstacle of safes built with copper layers interleaved among the layers of hard steel to dissipate the heat of an ordinary cutting torch. By feeding the oxygen through an iron pipe; the burning of the iron supplying sufficiently concentrated heat, they were able to gain entry to the safe. Steel mills soon acquired the technique and used an iron-oxygen lance to open plugged blast furnace spouts.

Progress IN REVIEW

PLASTICS

Plastic production is now about five times what it was at the start of the second world war. There is a slight recession in the industry at the present time because the supply no longer lags the demand, but it is anticipated that rapid growth of the industry will continue for the next decade. This optimistic attitude is held because new applications for plastics are being developed each day. The May 1947 issue of *Modern Plastics* gives a number of illustrations of the versatility of plastics. Polystyrene is being molded into plumbing fixtures and safe street light globes. Polyethylene watch straps are now grained to match alligator or pigskin and have many good physical properties. Acrylics are being used to make transparent, unbreakable dispensers for the canning industry. Many inexpensive plastics are used to make musical instruments with good tone qualities. Similar to these examples, is a molded correcting lens for use with projection type television sets, mentioned in the April 1947 issue of *Plastics,* 11 in. in diameter.

Silicone-glass laminate production is proceeding rapidly at the request of the U. S. Navy, and through the close cooperation of many laminators and suppliers of silicone resin. The production of these laminates follows intensive development for improved strength, non-toxic, high heat- and arc-resistance qualities. Another silicone-glass development program, reported in *Modern Plastics,* May 1947, is the perfection of chemically-resistant deck plates for use over the batteries in submarines. The Navy has already used the insulating properties of silicones in the production of lightweight generators and transformers for aircraft usage.

Plastic foams, having the properties of light weight, high strength-weight ratio, low thermal conductivity, and mold- and water resistance, are used as buoyant materials, as thermal insulators and as a core material for sandwich laminates used in the construction of luggage and airplane parts. Another core material, the honeycomb, until recently limited to flat structures, has now been commercially adapted for contoured laminates. The May 1947 issue of *Modern Plastics* outlines a method for manufacturing contoured cores for airplane wing sections by cutting them with a specially designed saw. The saw rides on a wooden base that gives the desired contour.

A plastic coating sprayed on polished stainless steel sheet is flexible enough to withstand severe forming operations. The coating protects the surface from scratching and acts as a lubricant in deep stamping operations. According to W. A. Phair, *Iron Age,* May 15, 1947, the coating is easily stripped from the pieces after they are formed.

Plastic screening, woven from extruded filaments of vinylidene chloride, is used in a number of ways because of its toughness, flexibility, chemical inertness, non-toxicity and long wearing properties. In some instances it is lacquered or metal plated; in others it is coated with cellulose acetate. The April 1947 issue of *Plastics* indicates that it is being used for screening, seat covers, filters of various types and window shades and awnings.

There are so many types of synthetic rubber and rubber-like resins that reviews are constantly necessary. The April 1947 issue of *Rubber Age* surveys the seven pliolites. The original prewar pliolite was cyclized natural rubber and was used extensively in acid- alkali- and water-resistant paints for steel, wood, concrete and paper, as well as in moisture-vapor proof heat-sealing paper coatings. Pliolite S-1 is cyclized synthetic rubber of a polyisoprene type and Pliolite S-2 is similar but produced with an emergency catalyst. These two types are used in the wire insulation field. Pliolite S-3, S-5 and S-6 are all butadiene-styrene synthetic rubber copolymer types. Pliolite S-3 is used for gaskets and wire insulation; S-5 is used for protective coatings and S-6 is used as a reinforcing agent in all types of synthetic rubbers and is used as a wire insulation.

Neoprene, because of its abrasion resistance, is being used in marine, water circulation pumps. First used on Naval units during the past war, gear driven, gear pumps, with neoprene impellers, survived many amphibious landings on the sandy beaches of the Pacific. The impellers were molded, and in many instances outlasted their metal housings. The decrease in size, caused by the wear, was counteracted by the few thousandths swelling of the neoprene after use.

Standard furnace carbon black is easily produced but is not quite as good as channel carbon black. However, Statex K, a new furnace carbon reported in *Rubber Age,* April 1947, has been shown to equal or surpass standard channel blacks in road wear tests on both natural and synthetic rubber tires. The new production process operates at twice the efficiency of existant channel methods and uses much less natural gas.

Natural rubber has been returned to free trade by recent Congressional action. However, the Government will still have authority over the manufacture and sale of synthetic rubber. The price of natural rubber is expected to go to a higher point while the price of the synthetic will continue to come down. Government action has also eliminated the restrictions on the production of white sidewall automobile tires, but shortages of necessary raw materials will delay quantity production for some time.

Tires and inner tubes were made almost exclusively from natural rubber before the past war, and the general acceptance of synthetic rubber in this field was rather gradual. Tires made from synthetic rubber of the GR-S type were long considered to be inferior; inner tubes made from synthetic rubber of the

butyl type were soon proven superior to those of natural rubber. The GR-S type was constantly improved until tires made of that material were as good as tires made of natural rubber. . . . in all instances except for tires in heavy military uses. Recent tests have indicated that neoprene, when substituted for a part or all of the elastomer in conventional tires, will provide definite improvements. The greatest advantage is obtained by designing a tire to make use of the special properties of the neoprene. Neoprene sidewalls or sidewall veneers offer an easy method of improving tire quality and appearance at no extra cost. Data obtained from over 10 million test tire miles, as reported in the April 1947 issue of *Rubber Age,* indicate that neoprene tires are better in many ways than both natural and GR-S tires.

Thin films of ethylcellulose greatly improve the resistance to breakage of glass, ceramics, plaster and wood. The film prevents breakage not only because of its own toughness but also because it distributes the shock. In the sports world it has doubled the life of bowling pins; in the home it has made vases and plaster figurines more durable. Similarly, the May 1947 issue of *Pacific Plastics* reports that a polyester type coating has extended the life of plywood forms for concrete construction. The coating improves the resistance to weather and concrete and retards delamination of the plys. Reapplication of the coating is simple but is not necessary until after 20 to 30 uses.

Striped or multi-colored thermoplastic products can now be extruded in one operation through a single die head. As reported in the May 1947 issue of *Modern Plastics,* strips, tapes, coils and wire coverings have been produced. This technique can be used for decoration and identification.

Cords needed for long wearing automobile tires, as well as the rubber and coloring compounds, are being developed through continual research. Cotton cords were replaced by rayon cords, or a mixture of the two, some years ago, and these are likely to be replaced by glass cords. Glass fibers are high priced at this time which will hinder the conversion process. Meanwhile, one large tire manufacturer is investigating the use of both wire and nylon for the cords of tires. Tires with wire cord have promising possibilities for off-the-highway use, such as is needed by tractor trailers. *Rubber Age,* April 1947, mentions the inherent advantages of excellent heat dissipation, high strength under high temperatures and little stretch under heavy loading. Precautions for best performance of these tires include: Operation at pressures 30 percent higher than used for similar sized cotton or rayon corded tires; and, installation in pairs because the wire cord tires do not "grow" in service as do ordinary tires. As reported in the April 1947 issue of *India Rubber World,* nylon corded tires have advanced to the commercial stage. The nylon is being used in the tread and increases tread life.

IRON AND STEEL

Ordinary soft steel was cold-extruded and cold-drawn to high degrees of reduction, without intermediate annealing, by the Germans. Steel cartridge cases were made in large quantities by this process. A lubricant retaining, phosphate coating is applied to the exterior of the billet so that the extrusion pressure or drawing tension goes into deformation of the steel instead of into die friction. W. W. Galbreath reported very favorably on this process to the U. S. Department of Commerce.

Compound metal plates were produced in large quantities by the Germans during the last war. Stainless steel surface sheets were hot-roll welded to plain steel backing plates. According to J. C. Richards, Office of Technical Services report PB-52882, copper, silver, brass and other metals were also used for surface sheets. In this process the plates were built up in a pack; the pack was sealed by welding, and piping was installed to pass a hydrogen atmosphere between the metal surfaces. After heating to rolling temperature, the pack was rolled in an ordinary plate mill. The easily fabricated plates had a combination of high strength with resistance to chemical attack, and could be made to have magnetic properties and an austenitic surface layer. The surface sheathing averaged between 1 and 2 mm. and the total plate thickness was about 1 centimeter. Penetration of a test plate by a fragment of shrapnel failed to separate the laminations.

Silico manganese spring steel, boron treated and produced with controlled hardenability, was used in making torsion bar springs for Hellcat Tank Destroyers. The metal was heat-treated in bars of 2.15 in. diameter, and a yield strength of 200,000 lb. per sq. in. was found in a test specimen taken halfway between the center and the outside. The mill-finished bars were annealed for machinability and, since high silicon steel decarburizes readily, the bars were machined to a diameter of 1.90 in. to get below the decarburized area. As mentioned in the May 1947 issue of *Metal Progress,* the bars were then ground to a surface finish of 70 micro inches or better, magnafluxed, heat-treated, and magnafluxed again. The hardenability of the bars was carefully controlled to produce a compressive stress in the fully hardened outside layer. To get this condition the interior of the bar can not be completely hardened. Finally the heat-treated piece was shot-peened, and preset by twisting. This increased the compressive stress at the surface and slightly increased the yield strength. Standard automobile leaf springs are not manufactured with all these precautions and do not get high static and fatigue strengths. Decarburization is not removed and the final fatigue strength is about half what it would be if the leaf springs were made as these torsion bars are made.

Nailable steel flooring, designed to lower maintenance costs for freight handling equipment, is made of steel channels 8 in. wide. The channels are laid across the width of the car and are welded to the underframes, with a small space between each channel. Nails driven into the concentric flanges of curved steel are deformed and held more securely than in wood. The grooves are filled wth a self-sealing plastic composition that prevents loss of bulk freight. Since the steel channels have a greater structural strength than previously used wooden flooring, the underframes for the carriers need not be as heavy.

The use of high frequency induction heating for annealing, softening, tempering or strain-relieving small steel parts or sections of parts has been carefully studied by the U. S. Army. The results of this research are described in the report PB-49265 of the Office of Technical Services. It was found to be possible to change the physical properties of a section of a steel piece without affecting the rest of piece, because it is possible to concentrate and localize high frequency induction heating on a small area. High frequency induction heating is also useful because it: Reduces operating time with its high heating rate; uses equipment that needs no preheating; produces uniform results and forms less scale on steel parts. A thin rim of a steel gage has been successfully tempered by induction heating.

Atomic hydrogen welding of thin austenitic steel jet engine combustion chambers results in crack formation when done with a flux containing borax or borates. The hydrogen reduces the boron from its compounds, and the boron alloys with the weld metal to produce high brittleness. As mentioned

by G. Richardson in the May 1, 1947 issue of *Iron Age,* the use of an ordinary hydrogen jet, to shield the joint during welding, will counteract the undesirable reducing action.

Acceptance testing of high speed tool steels for the U. S. Navy, by actual cutting tests, is discussed by W. H. Wooding in the A.S.M.E. *Transactions,* May 1947. The metalographic structure of the steel was found to be the most important factor in tool life. Red hardness increases with a decrease in the amount and degree of agglomeration of separated carbide, and increased acicularity of the ground mass. It was found that for continuous roughing cuts, without coolant, tool life increased with increased red hardness. In interrupted cutting, with out coolant, where toughness is highly important, a metallographic structure in which the acicularity of the ground mass is just disappearing gives optimum tool life. These results are found for both tungsten and molybdenum tungsten types of high speed tool steels. The rate of solution of carbide in austenite during the heat-treatment specified by the steel maker is believed to vary according to the prior processing of the steel; and the degree of solution controls the microstructure. The Navy research indicated tool lives varying by 100 percent between supposedly like steels from different makers. It can be said that the annealed structure does not indicate what the performance will be but that the structure after hardening and tempering will so indicate.

Steels of different carbon levels, with like Jominy curves, have higher notch toughness when the carbon is low and hardenability is supplied by alloying elements. A special report, No. 36, on a Symposium on the Hardenability of Steels, held by the British Iron and Steel Institute, agrees with American thinking by stating that it is impossible to accurately calculate hardenability from the chemical composition of a steel. It was found that steels of the general type of the American N. E. series, using about 0.30 percent carbon, high manganese, fractional percentages of nickel, chromium and molybdenum, and intensified with boron, produced the most economical, deep hardening steels.

Supersonic flaw detection has been applied to large steel forgings and castings, and gives valuable information on the internal structure of irregular and regular shapes. A number of large steel forgings were echo examined; those with normal thermal records showed no cracks and those with abnormal records showed hairline cracks. A physical examination of the blocks proved that the supersonic indication was correct. As mentioned by H. Nicholson and T. Rotherham, *Steel,* Feb. 24, 1947, the use of a sound wave should be regarded as a new method of investigation capable of giving information about the internal character of large masses of steel, which would not be possible by methods now in use.

Stainless-clad steel is now a practical material for the fabrication of automobile bumpers, and solves the problems of pitting and flaking. Interest in this material was first aroused because of the shortage of electroplating equipment for chrome and nickel plating the conventional type bumper. The composite metal, having an outside layer of stainless steel backed up by a low alloy high strength steel, gives the ideal combination of shock resistance and lasting corrosion resistance, with toughness.

NONFERROUS METALS

Preparation of nickel flake for painting purposes, by mechanical methods such as hammering, ball milling, etc., is slow because of the toughness of the metal. The thinnest flakes produced by this method are about 0.00004 in. thick. Using 0.20 to 0.31 grams of thiourea per liter of solution, a British company has developed a plating process for preparing brittle, lamellar nickel. The nickel, or cobalt, is plated out on cathodes made of aluminum, tantalum or chromium alloys. The brittle, mirror-like, non-adherent plates can be broken up in a ball mill. Flakes as thin as 0.000012 in. will be produced after 90 hr. of milling. Treatment with dilute hydrochloric acid prior to grinding will give plates as thin as 0.000010 inches. Plates of half this thickness can be produced by use of interrupted current in the plating operation.

To reduce the losses in the average foundry due to unsound bronze castings, caused by poor metal practices, the use of a new type of degassing cover is recommended. For best results, the cover compound, 1 lb. of Cuprex and a degassifier made of 11.2 oz. of Foseco R6 per heat of 280 lb., should be added to the bottom of the furnace to protect the cold charge. An oxidizing atmosphere is developed, according to S. W. Wysocki, *Iron Age,* May 1, 1947, which protects the charge as it is added and melted. To make dense castings, the degassifier was added to the bottom of the preheated ladle, the metal was poured in, stirred, skimmed, and deoxidized with phosphor copper prior to pouring into molds. It is said that inexperienced help, using this method, have produced castings with better physical properties. 25 percent increase in tensile strength has been reported, improved density and at an added cost of 1 mil per pound.

Lead, not as spectacular as many of the other nonferrous metals, continues to be a useful and dependable material. Discussed in *Lead,* Vol. 17, No. 1, are a number of uses for lead that are not widely recognized. As a construction material, lead is being used to permanently rainproof buildings. Extruded lead cap strip will protect masonry joints from all kinds of weathering and is easily applied. Lead strip has been used to repair old buildings and to protect new structures. Because lead is highly resistant to corrosion, and also very flexible, it is being used as a roof flashing material. Cornices and pediments are completely inclosed by the lead sheets and tight joints can be made with the roofing materials. Entire installations can be made without using solder; all joints are loose locked and filled with non-hardening roofers calking compound. Where lead sheets are in contact with new masonry, they are coated with an asphalt paint to protect them from the free lime in the mortar. Lead and lead alloys are also used in the form of die-castings. One company, manufacturing fire extinguishers, uses valve housings made of die-cast lead. In this instance, the weight of the lead is used to advantage and the stability of the lead insures the effective operation of the ball check valves after long periods of idleness. The principal field of lead alloy castings is as a corrosion resistant covering for steels. The lead takes and holds paint well and also protects against atmospheric moisture. Hot dip lead alloy coatings are not limited to steel. They can also be used on cast iron and brass fittings. A lead coating, heat-bonded to cast iron, acts as a tinned surface when parts are to be soldered. The lead coating provides corrosion resistance and increases the efficiency of the joint.

Standard magnesium extrusions are being extensively used now that the price for them has come down to within 5 cents per ft. of the cost of similar aluminum shapes. At least one company now stocks standard magnesium shapes and also maintains dies for extruding special shapes. The actual cost of these shapes is less than the price differential indicates because they save machining time and weight in the finished product. As mentioned by H. Chase, *Materials and Methods,* April 1947, extruded magnesium is used for the vibrating members of door chimes, and is used in organ chimes and musical instruments

where tonal qualities are important. To eliminate the scuffing caused by aluminum parts sliding on each other, extruded magnesium sections are being used for window sash and casements. Scuffing does not occur between aluminum and magnesium.

Colorimetric determination of iron in light metals has been developed by the Germans. As outlined in the Office of Technical Services Bulletin, PB-53890, the light metal alloy is dissolved in an acid solution. This solution is then combined with specially prepared sulphosalicylic acid. The presence of iron causes the solution to turn red and the amount of iron present can be measured with an electric colorimeter using a green filter. This method is said to work satisfactorily in the presence of Al, Mg, Si, Cu, Zn, Pb, Cd, Sb, and Mn without the necessity of separating the iron.

Elgiloy, a recently developed spring material, is described in the April 1947 issue of *Materials and Methods*. The alloy is made up of: Cr, 20; Co, 40; Ni. 15½; Fe, 15; Mo, 7, Mn, 2; C, 0.15 and Be, 0.03 percent. Elgiloy was developed primarily as a watch spring material that would retain its full strength over a long period of time, reduce spring breakage and help watches retain their original accuracy. The alloy is ductile and can be worked hot or cold. The reported mechanical properties for this alloy are generally comparable to those of high carbon spring steel. Elgiloy is not magnetic at any time and resists corrosion very well. With an increase in production of the alloy, many uses will be found for it in the fabrication of other parts.

Hydrogen can be effectively removed from molten aluminum, according to an article in *USCO News Casting,* May 1947, by passing nitrogen or chlorine gases through the melt. These gases must be free of moisture and can be introduced into the melt with a small graphite tube with the open end at the bottom of the pot. Iron tubing can be used for nitrogen if coated with a suitable refractory. With a melt temperature of about 1,300 deg. F., the gas flow should be just enough to agitate without splashing. For a 250 lb. melt, the gases should flow for a period between 5 and 10 minutes.

Die-casting equipment of the "cold-chamber" type, almost universally used on aluminum, magnesium and copper base alloys, has not replaced the gooseneck machines for casting zinc base alloys. Molten metal must be hand ladled from the holding furnace into the well of the casting machine in the cold chamber process, which reduces output. An automatic feed attachment for the cold chamber machine, described by H. K. and L. C. Barton in *Mechanical World,* Feb. 14, 1947, injects molten metal directly into the well. The new feeding device, termed a "hot chamber", uses a motor driven Archimedean pump to lift the metal from the holding furnace to a well in the pressure chamber. On machines with a capacity from 12 to 14 lb., designed for casting aluminum, magnesium and zinc, the cold chamber is on one end and the hot chamber is on the other end.

Precision plating of bearing alloys can now compete with present methods on a cost basis through use of recently developed methods. Precision plating will also give better results because the bearing dimensions can be accurately controlled. Maximum adhesion is had when the plate is put on anodically etched base metals. Silver is plated on ferrous base metals; and lead, indium or lead-tin-copper are plated on silver and copper base metals. The adhesion can be further increased by heating the bearings in an oil bath from 320 to 340 deg. F., for a period of one hour. As mentioned by R. A. Schaefer, *Iron Age,* April 10, 1947, the first plate may be overplated with a ternary deposit of lead-tin-copper. Plated tri-metal bearings are becoming standard for medium and heavy duty truck, diesel and aircraft engine applications.

Expensive hand buffing ordinarily required for electroplated silver has been largely eliminated in a process described by D. Gray and S. E. Eaton in the April 10, 1947 issue of *Iron Age.* The racked parts are removed from the plating bath and, without any intermediate rinse, placed into an electro-polishing bath with the current reversed. This dissolves the high spots and leaves the parts with a mirror-like finish. General practice still indicates an alundum color buff for the final operation.

Magnesium alloy brick tongs, weighing 2¼ lb., are replacing conventional steel tongs that weighed 6¼ pounds. Another item to lighten the burden on labor, also reported in the April 1947 issue of *Light Metals,* is a 1 lb. 11 oz. plasterer's "hawk." Magnesium alloys have shown that they could stand the hard treatment these tools receive and their light weight has justified the added cost. A similar example of magnesium alloys replacing long-used materials appears in the same issue. Magnesium alloys are now being used to replace copper and zinc for photo-engraving plates. The standard FS-1 alloy is used, as is a modification with lower calcium and magnesium content: Al, 3.0; Mn, 0.3; Ca, 0.2 and Zn, 1.0 percent. The FS-1 is used for halftone work and the modified alloy is used for line work. Magnesium alloys etch faster than copper and can be used for finer line work than zinc. Such plates are lighter to handle, etch evenly and can be subjected to the same shop processing as that used for the standard zinc plates except for the plate preparation and the strength of the nitric acid used.

Inert gas welding of aluminum and magnesium alloys and stainless steel can be done on sheets of thicknesses ranging from 0.025 to 0.500 inches. As stated by H. R. Clauser, *Materials and Methods,* April 1947, argon has replaced helium in this process because it is more abundant and because it is heavier and shields the puddle more effectively. When argon is used, all metals can be welded without the use of a flux.

Electrolytic zinc, 99.999 percent pure, can be recovered from pyrite cinders by an amalgam process first developed by the Germans. The process has been used commercially and, as indicated in the *Daily Metal Reporter,* May 1, 1947, free chlorine is one of the industrially useful byproducts of the process.

High pressure aluminum alloy die-castings are more economical than sand-cast iron parts for use in some light-weight gasoline engines. While the aluminum die-castings are slightly more expensive than sand-castings, savings in subsequent machining operations and set-up time are reported. R. P. McCullough, *Iron Age,* May 15, 1947, indicates that die-castings reduce the need for specialized machinery and enable model changes to be made more easily.

Electroplated parts of high quality have been produced by a periodic reversal of the plating current at short intervals. Unsound metal deposited previously was deplated and a greater homogeneity was obtained than in deposits produced by continuous methods. For most applications the process uses a plating cycle of 2 to 40 sec. with a reversal period of ½ to 5 seconds. As mentioned by G. W. Jernstedt in the April 28, 1947 issue of *Steel,* the reversal removes from 10 to 50 percent of the metal deposited. The surface of the metal is smoothed and brightened by each reversal, and it is claimed that almost any thickness can be produced without flaws. The current reversal reduces the polarization of the solution and speeds the plating part of the cycle. The method has worked well with copper and has been successfully used with silver, zinc, cadmium and gold.

Vibration Testing Technique And Its Use in Improving Designs

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Shake testing to check the dynamic behavior of machines and structures at operating conditions and thus obtain reliable data for use in improving and developing the design of a product. The need for such data is described. A brief summary is included outlining forced vibration methods for determining the fatigue strength of individual parts.

TO OBTAIN data that will assist engineers in improving the design of their products, many testing techniques have been developed. With suitable equipment, testing will reveal to the engineer much that cannot be calculated concerning displacements, deflections, reactions, vibrations and resonances, and their intercorrelation, that occur over a range of speeds in structures and mechanical equipment.

The need for such data is made obvious by many common experiences. Many structures, apparently proved to possess ample margins of safety when loaded statically to destruction, are found in operation to crack at joints or at sections that change abruptly. Proper testing would tell whether the cracks were caused by resonance of the structure in response to a dynamic excitation, or by some member yielding under a static load thus inducing a redistribution of the load. Testing would also indicate whether relatively low repetitive stresses in service might cause failures at loads well below the designed static yield load.

Data concerning the distribution of loading and deflections of important members are necessary so that fatigue limit and stress concentration factors can be applied with some assurance.

From vibration tests the engineer can learn when the speeds of operation approach the mechanical resonance of parts, that is, when the frequency of an exciting force is approaching the natural frequency of a mode of vibration. Under such a condition, the magnitude of the resulting dynamic forces are not controlled by strength (short of failure) but are limited by the internal damping of the system.

Fig. 1—Exciting an airplane wing with a vibration exciter to check structural flexibility and to reveal any excessive stress concentration.

Damping present in most fabricated steel structures will allow an amplitude of vibration to build up at resonance to approximately 200 times the amplitude of the exciting force. This build up can be reduced to about 20 times that of the exciting force by the use of a material having inherent damping qualities, such as rubber.

Theoretically an ideal vibration isolation system should locate the resonant frequency of all modes below the operating range. But practical limitations frequently tax designers to achieve this condition. Further, isolation is not 100 percent effective. If 10 percent of the disturbing force is allowed to pass through the isolator units to excite a resonant structural mode having a build up of 100 to 1, the resulting amplitude of vibration will still be 10 times greater than that of the disturbing force. Such a resonant mode is not easily attacked by improving the effect of the first set of isolator units, but will respond to proper application of damping material or changes in the natural frequency of the offending structure.

To design most structures to predetermined resonant frequencies of all significant modes is not simple. The calculations necessary resemble

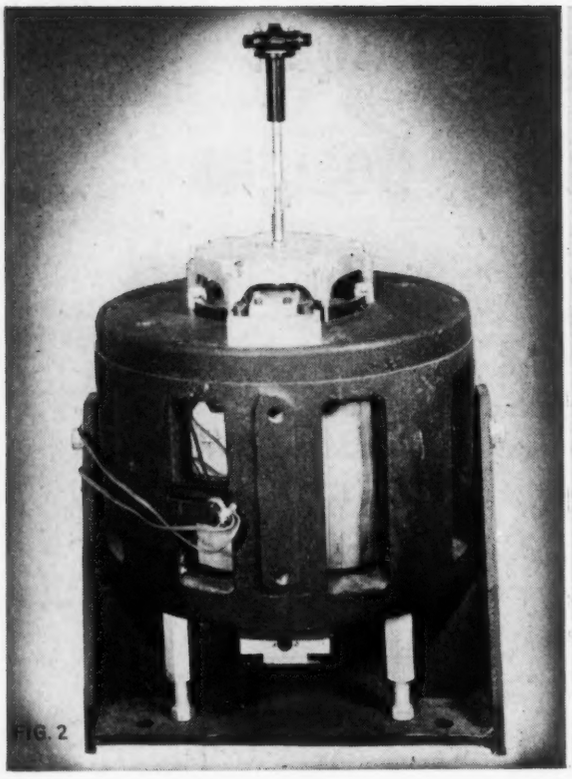

FIG. 2

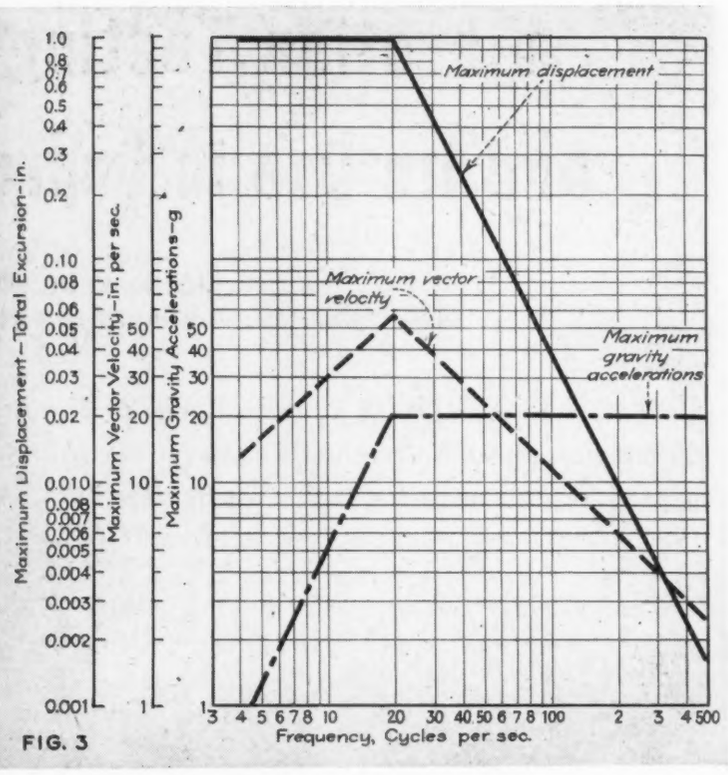

FIG. 3

Fig. 2—Small gear mechanism mounted firmly as a mass on the table of an S3 electro-magnetic vibration exciter. The test can be limited to a standard acceptance test, or the mechanism can be vibrated over a suitable frequency amplitude range to locate possible service difficulties.

Fig. 3—Displacement, velocity and acceleration curves over a frequency range of S3 exciter having a 10-lb. moving system and 200-lb.-rated force.

Fig. 4—Test setup and analysis of vibration in a free-free beam system. (A) Arrangement of exciter and beam to demonstrate vibratory characteristics. (B) Resonant modes and corresponding frequencies of beam. (C) Vibratory response of beam showing the effects, and their relationship, of changing the frequency of the exciting forces on a simple structure.

aircraft flutter computations. In practice the designer is limited to strength checks of a few sections of simple geometrical shape. Deflection calculations of most elements under load are more frequently omitted. To determine natural frequency, the stiffness of the element is usually checked with its weight distribution. A few parts, such as crankshafts, have been improved by mathematical analysis methods, but these methods are too complicated for wide usage. But even after the engineer has done his best to design a smoothly running machine, the quality of its performance is unknown until tested.

In the manufacture of the machine, available materials may have been substituted for those specified, fabrication methods may have been changed and new machine operators may have been used. This practice, though justifiable, increases the unknowns of residual stress, assembling stress, stress concentration factors, and fatigue strength of the materials.

In addition, unfortunately, in many products there are unknown factors that contribute to noise and vibration, but each is so masked by others that none is neither easily analyzed nor observed during normal operation of the machine.

To secure reliable data that can be used in improving and developing a product design, simple tests and checks of the dynamic behavior of a machine or structure can be made by shaking the machine with a vibration exciter and then studying the resulting motions of parts of the machine with pickups or stroboscopic light. If desired, the structure can be resonated to destruction.

Rotary vibration exciters can be used to excite structures and special mechanisms linearly or in a plane. This type of equipment usually applies a small force at low speed, with forces increasing with speed. While this equipment has been widely used and is effective, it is usually fairly complicated to apply. Also it requires adjustment for each force desired and is limited in frequency range.

Electro-magnetic vibration exciters have been developed for wide frequency coverage with independent continuous control of a pure sine wave force. These units have moving elements that are extremely light in weight. While the application of this type of equipment is limited to systems responding to a linear sine wave force, with a little ingenuity a torsional or other system can be devised by the test engineer to meet special test setups.

Small assemblies or parts firmly mounted on the table of the electromagnetic exciter, as shown in Fig. 2, can be vibrated with an acceleration proportional to the force developed by the exciter and inversely proportional to the weight of the moving system. Fig. 3 shows the amplitudes to be expected at different frequencies when shaking with the vibration

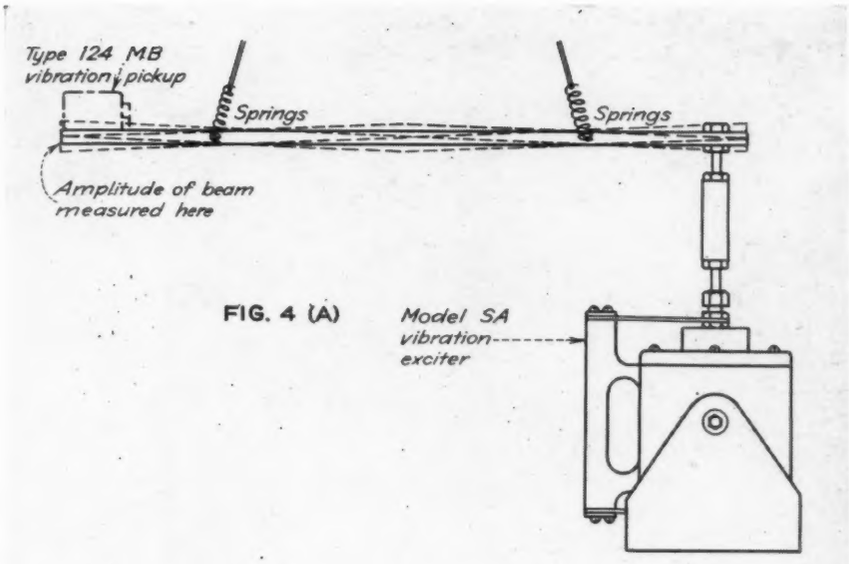

FIG. 4 (A)

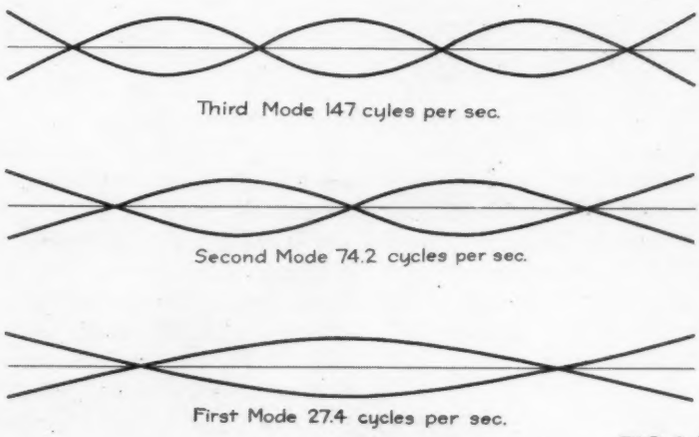

FIG. 4 (B)

obvious design change to strengthen the part. Any design improvement that can be made while parts are still in production, of course, is desirable. But the significant value of such testing is the knowledge gained of inherent weaknesses that can be avoided in future designs.

The free-free beam and vibration exciter shown in Fig. 4 demonstrates the resonant modes of vibration and the amplification experienced in each. In this system, Fig. 4(*A*), the driven part of the exciter becomes a part of the spring mass system. The force delivered by the exciter is kept constant and the frequency is increased while the amplitude of the beam is measured. In Fig. 4(*C*) the response of the beam is plotted as displacement, velocity, and acceleration as ratio to gravity acceleration to show their relationship.

The displacement-frequency graph Fig. 4(*C*) shows that:

1. At low frequencies, the system follows the displacement to be expected by an equivalent mass.

2. At point *A*, the system breaks away from the mass characteristics and shows a typical resonance curve to point *C*.

3. The peak response *B* of the first resonant mode occurs when the mass is 180 deg. out of phase with the restoring force, under which condition

exciter shown in Fig. 2. Pickups can be calibrated by this method.

Large assemblies and machines, supported on floors or benches, can be tested by attaching the vibration exciter to any heavy component part of the machine, such as a frame, bearing housing or an integrally mounted electric motor. In testing, the exciter vibrates the machine under test at its resonant frequency in each significant mode. From displacement observations, sections of parts can be checked for stress concentration to determine if design changes are required. When necessary, brittle coatings or wire resistance strain gages can be used to evaluate the stress.

Either the amplitude or time can be increased to produce a typical operating failure. Observation of a fatigue failure usually indicates an

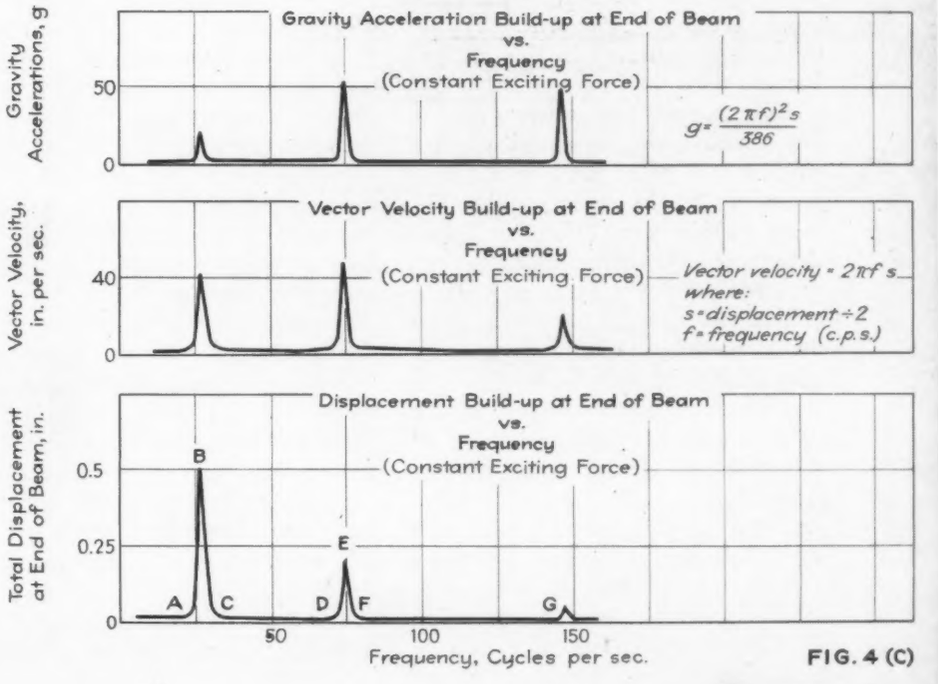

FIG. 4 (C)

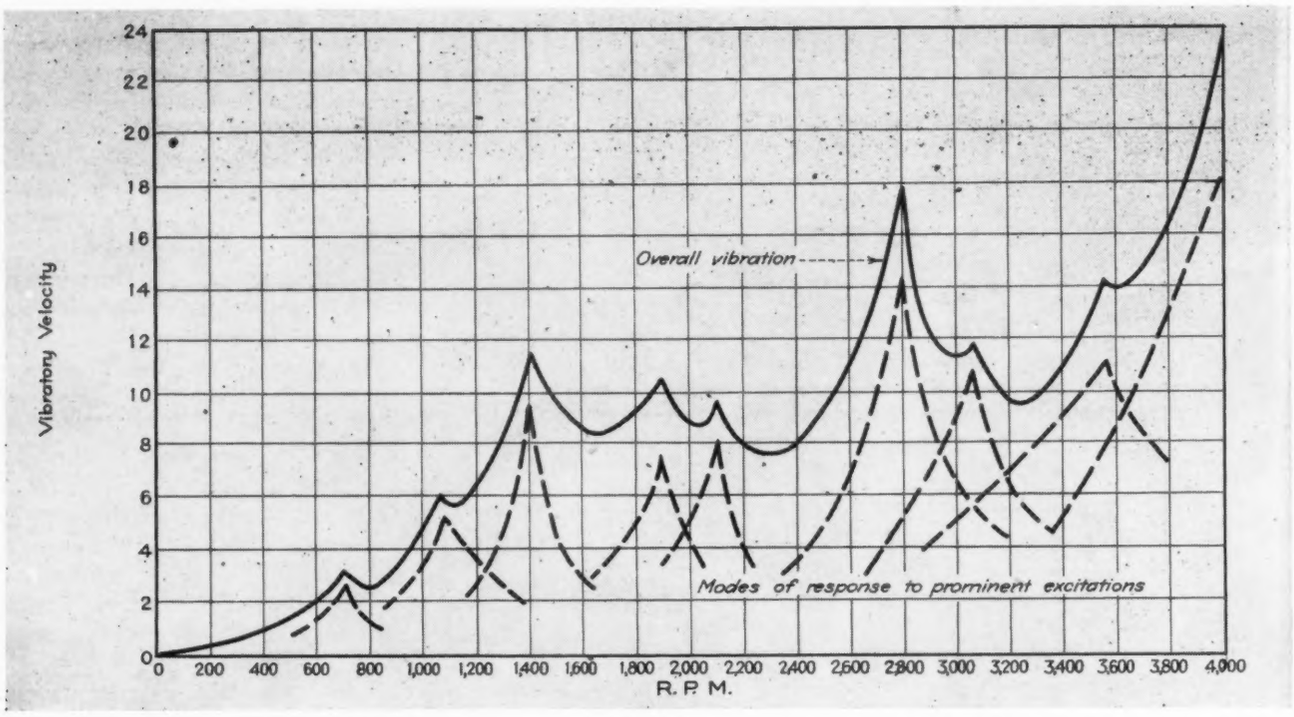

Fig. 5—Major modes of response to the prominent excitations contributing to a high overall vibratory level.

the amplitude builds up until the damping energy expended per cycle is equal to the driving energy supplied and at that time the driving force is 90 deg. out of phase with displacement.

4. Increasing the frequency above that of point *C* decreases the displacement as expected from the spring mass system to point *D*, where a higher mode becomes predominant and the system repeats its resonant curve to *E* to *F*. At *G* the third mode becomes prominent. Higher modes are neglected in this analysis because they are insignificant.

Determining the behavior of a beam with a single excitation and a single phase of response is a simple procedure. The problem becomes more difficult when it is required to determine optimum channel sections to avoid resonance from one of the exciting forces originating in a direct driven motor generator set. Trouble may arise from any order of excitation that develops a response in any of the significant modes.

Fabricated structures usually have many significant modes of vibration, several of which are usually responsive to each order of excitation present in the operating unit. The resulting vibration spectrum usually resembles Fig. 5, where each excitation and its accompanying responses run together to make a high overall level of vibration and noise. The discouraging part of trying to improve such a structure is the amount of test work and analysis required to correct one excitation or mode, which when corrected disappears and leaves a high overall level.

The obvious thing to do is to summarize the excitations and the operating range of each excitation, then move the resonant frequency of all responsive parts of the structure out of the operating range. Where this procedure is not possible, the best compromise is to group the responses in an infrequently used portion of the operating range and add damping to the system until the results are satisfactory. It is well to keep in mind that most excitations, particularly rotating unbalance, have less energy at low speed and therefore resonances are usually less severe in the low range.

Another suggestion is to make most of the machine elements as rigid as possible, adding flexible elements of known characteristics at points as required to prevent resonance. Shortening the rigid members increases their frequency. The flexible materials generally add appreciable damping to reduce high frequency vibration and take the ring out of the structure.

A complicated problem to solve by mathematical analysis methods would be the reduction of noise in an air conditioning unit that consists of an a.c. motor, compressor and a four-blade fan, the respective speeds of which are 1,750; 600; and 1,150 r.p.m. The motor excitations are first order, the motor also has a hum of 3,600 c.p.m. The compressor has first order unbalance, second order unbalance, and third order torque impulse. The fan has first order unbalance and a fourth order air blast. The frequencies of resulting excitations on the system are therefore 1,750; 3,600; 600; 1,200; 1,800; 1,150; and 4,600 cycles per minute.

Shake testing the air conditioning unit with a vibration exciter over a range of frequencies revealed that minor rattles occurred at 1,400 c.p.m., which could be corrected by assembly technique. The test also revealed compressor rocking at 2,200 c.p.m.; cooling coils vibrating at 2,400 c.p.m. and motor support flexing at 3,200 c.p.m. to allow a torsional mode of the motor; these three defects could be corrected by mounting these units solidly on a sub-frame that could be shock mounted from the main frame. (See A.S.M.E. paper "A Simplified Method for the Design of Vibration Isolating Suspensions" by R. C. Lewis and K. Unholtz.) Panel flexing was observed at 4,400 c.p.m.; which could

be corrected by forming the panels or using a heavier gage of metal.

Vibration testing has shown that mechanisms such as tractors, which use the engine as a part of the frame, should have driver components such as seat and steering wheel isolated. Also that radiators and fenders should be either stiff enough to avoid resonant build up or be isolated to prevent destructive vibration with its accompanying noise.

Many parts of tractors such as fenders have such severe resonance at operating frequency that dust gathers at mode locations. While such patterns can be used as a basis for design changes, the data obtained by the following procedure will be more helpful in giving designers a much better understanding of the vibrations present and the effective means for their correction.

1. Draw a vibration spectrum as shown in Fig. 6 that indicates the significant excitations and locates the idling and operating speed ranges.

2. Attach the vibration exciter to some point of the tractor to simulate each significant excitation, such as vertically on the frame on one side of the engine to simulate cylinder firing excitations, or vertically on a wheel to simulate traction lug frequency. Response of fenders to a high frequency gear tooth excitation can be simulated by attaching a light weight vibration exciter to the fender supporting frame.

3. Scan the frequency range and note the frequency and mode of each response.

4. Make the changes necessary to get each response as far as possible from its exciting frequency.

5. Recheck the frequency response to determine the effectiveness of changes.

6. Check any modes that are considered significant for stress concentrations and possible fatigue.

In addition to the testing of complete assemblies, vibration exciters can be used to determine the fatigue strength of standard test specimens and individual parts. Torque shafts, push-pull rods, frame members and many other parts can be fatigue tested by the addition of mass to produce the desired load at a specific frequency. The following is a brief outline for testing such parts.

Torque Shaft. Add a mass at each end of the shaft that is roughly equivalent to a circular disk 10 to 15 times the diameter of the shaft and of thickness approximately equal to the shaft diameter. Attach the vibration exciter to drive tangentially at either mass.

Bending can also be excited in the shaft by driving parallel to the shaft at a point near the circumference.

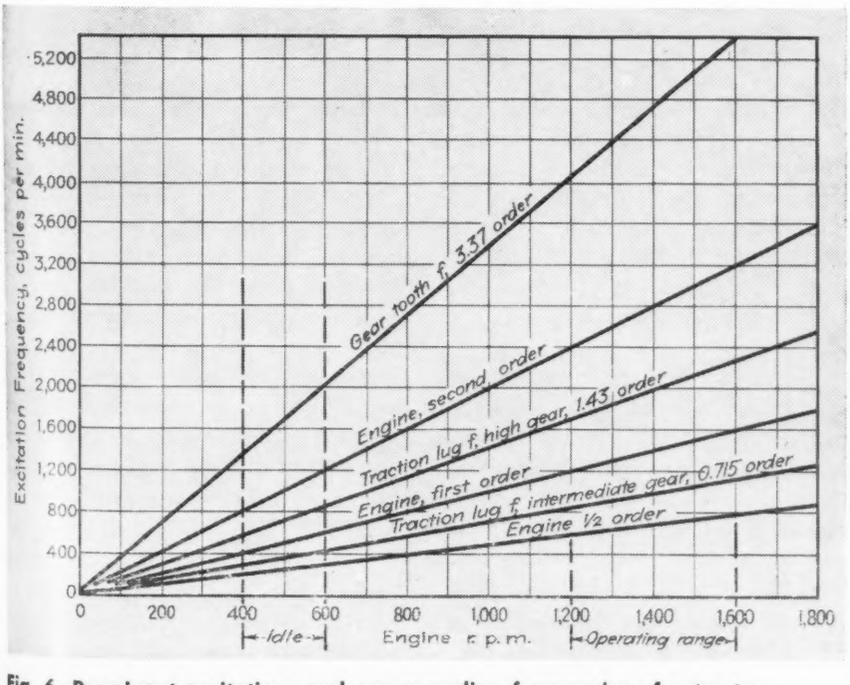

Fig. 6—Prominent excitations and corresponding frequencies of a tractor.

Push-Pull Rods. Add a mass to each end of the rod and drive from either mass on a line coincident with the centerline of the rod. Deflections in rod will be relatively small, it may be well to add a spring in series with the rod to increase motion. Another arrangement, to give a positive or negative answer, is to use a necked down section of known material in series with the test specimen.

Frame Members. Add masses at locations to simulate concentrated loads. While this can be worked into fairly complicated setups, it is generally more satisfactory to simulate simple beam bending with concentrated loads at the ends and either a single load or symmetrical loads at the center.

Small Assemblies and Controls. Mount either directly on the vibration exciter table as shown in Fig. 2, or on a resonant table driven by the exciter and drive through a standard comparative test. The standard test used should simulate both the expected frequency and amplitude that the assembly will have in service.

In conclusion, vibration control is largely concerned with the natural or resonant modes of vibration and their relationship to exciting frequencies. The resonant frequency can be increased by adding stiffness or reducing mass, but not by increasing mass and stiffness in the same proportion. Resonant frequency can be decreased by adding mass or reducing stiffness, such as the addition of resilient gaskets or vibration isolators.

Vibration exciters allow the average engineer an opportunity to visualize the inherent modes of vibration of a structure and thus obtain a better understanding of the effects of vibration. The testing technique is a simple, direct approach to the reduction of noise and vibration.

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Fig. 1—Machines of special design are used to mill the ball races in Bendix-Weiss universal joints.

Constant-Velocity Universal Joints

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Bendix Products Division, Bendix Aviation Corporation

Fundamentals underlying the features and design of the Bendix-Weiss universal joint. The theory of the Cardan universal joint is referred to as the background to explaining the constant-velocity principle of motion transmission.

FOR centuries design engineers have used the Cardan universal joint despite the fact that it does not transmit power with uniform angular speed when the shafts make an angle with each other. The typical Cardan joint has a cross-shaped central member to which yokes are attached to connect the driving and the driven shafts.

Its construction causes the driven or follower shaft to have varying instantaneous rotative speeds although the driving shaft rotates at constant angular speed. The magnitude of this variation in rotative speeds depends upon the angle between the shafts, being greater as the angle β between the shafts becomes larger.

The curves of Fig. 2 show how the angular motion of the follower shaft varies from the angular motion of the driver for three angles between shafts. This cyclic variation occurs twice per revolution. The table of Fig. 2 contains data on variations in percent for several shaft angles.

Cardan Joint Analysis

A review of the Cardan joint theory is helpful in understanding the basic principle of the constant-velocity universal joint. The Cardan joint consists of a driving or primary shaft P and a driven or a secondary shaft S, each ending in a fork or yoke, Fig. 3. The two primary fork ends A-A' are connected to the two secondary fork ends B-B' by a cross shaped connecting link.

Let primary shaft P and secondary shaft S be displaced by the angle β in the plane formed by the P and S axes. Then one full revolution of P will obviously result in one full revolution of S. But any intermediate angle of rotation θ by the primary shaft P will not produce an identical change in the angle of rotation ϕ of the secondary shaft S. This well-known relationship of the displacement angle β and the angles of rotation θ (for P) and ϕ (for S) is a problem in spherical triangulation expressed by the Equation

$$\tan\phi = \tan\theta \cos\beta. \qquad (1)$$

Differentiating this equation to obtain the time rate of angular change, or angular velocity, results in the follow-

ing relationship between $d\theta/dt$, the angular velocity of P, and $d\phi/dt$, the angular velocity of S.

$$\frac{d\phi}{dt} = \frac{d\theta}{dt} \frac{\cos \beta}{1 - \sin^2 \beta \sin^2 \theta} \qquad (2)$$

Thus it is shown that even with constant angular velocity and torque (constant power) of the primary shaft P both the angular speed and torque of the secondary shaft S fluctuate as a sinusoidal function of β, the angular displacement of P and S, and of θ, the angle of rotation of P. Fig. 2 illustrates the magnitude of the fluctuation.

Homokinetics

In high speed or high angle applications lack of uniformity in power transmission has detrimental effects. As an example, steering knuckles of front drive vehicles with high working angularity β require a universal joint design that is homokinetic, that is, one without fluctuations of speed and torque since these would set up undue stresses in all component parts of the driving mechanism. Referring to the sketch of a Cardan joint Fig. 3, it can be seen that the four trunnions of the cross shaped connecting link engaging the fork ends *A-A'*, *B-B'* are the power transmitting elements, which will be referred to by the abbreviation *"PTE"*, between P and S. These four PTE are all located in one plane—the PTE plane—which, during one full revolution of the Cardan joint, changes location in space four times, alternating between a position perpendicular to P and perpendicular to S. This fluctuating in space of the PTE plane is directly responsible for the lack of uniformity in power transmission—termed "lack of homokinetics"—as shown in the analysis of the Cardan joint.

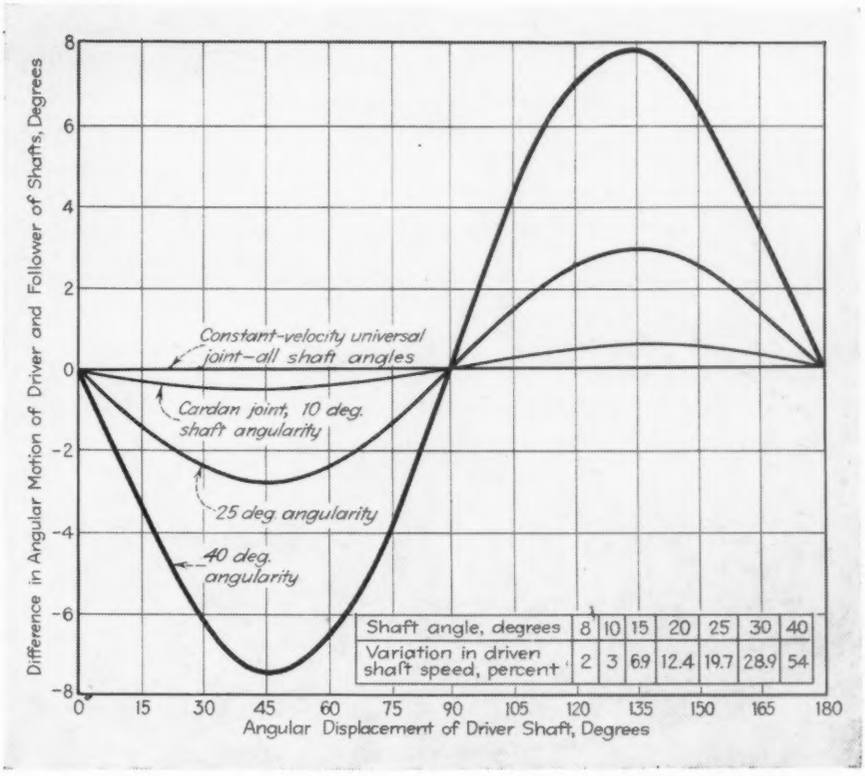

Difference in Angular Motion of Driver and Follower of Shafts, Degrees
Angular Displacement of Driver Shaft, Degrees
Constant-velocity universal joint—all shaft angles
Cardan joint, 10 deg. shaft angularity
25 deg angularity
40 deg. angularity
Shaft angle, degrees | 8 | 10 | 15 | 20 | 25 | 30 | 40
Variation in driven shaft speed, percent | 2 | 3 | 6.9 | 12.4 | 19.7 | 28.9 | 54

Fig. 2—Curves showing how the difference in angular displacements of driver and follower shafts in universal joints vary with the angularity of the shafts.

The mathematical prerequisite for homokinetics is that the PTE plane must be held constant in space, perpendicular to the plane determined by the axes of primary shaft P and secondary shaft S and bisecting the angle (180 deg. $-\beta$) between P and S, caused by displacement β of S from P, Fig. 4. This plane will be referred to as "homokinetic plane" which, in technical literature is sometimes loosely called "working-angle bisecting plane".

The necessary condition for a

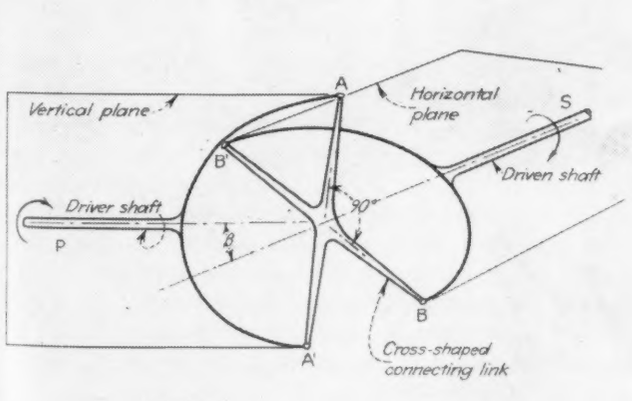


FIG. 3

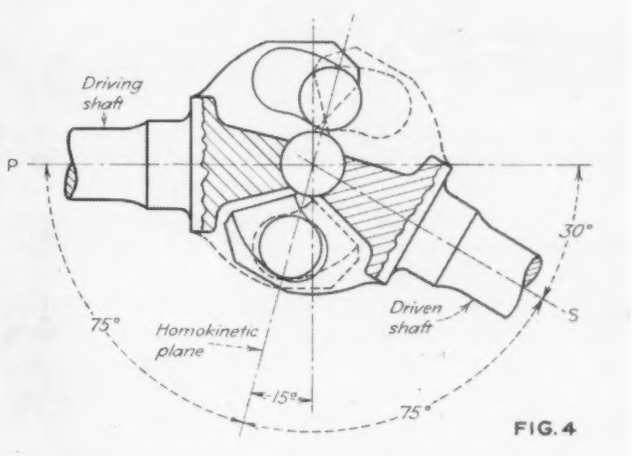


FIG. 4

Fig. 3—In the Cardon universal joint the driver Yoke A-A' rotates in a plane oblique to that in which the driven Yoke B-B' rotates, thereby causing non-uniform angular displacements of the driven shaft. Fig. 4—In the Bendix-Weiss universal joint the driving and driven contacts occur in the plane that bisects the shaft angle so as to transmit identical angular displacements from one shaft to the other.

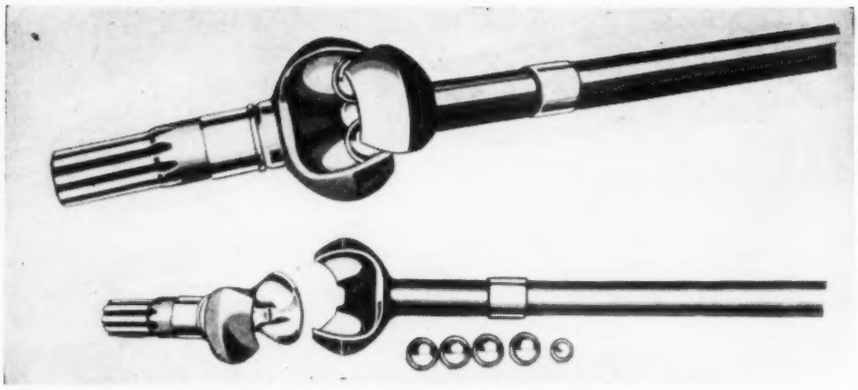


Fig. 5—Assembled and disassembled parts in a high angle type universal joint.

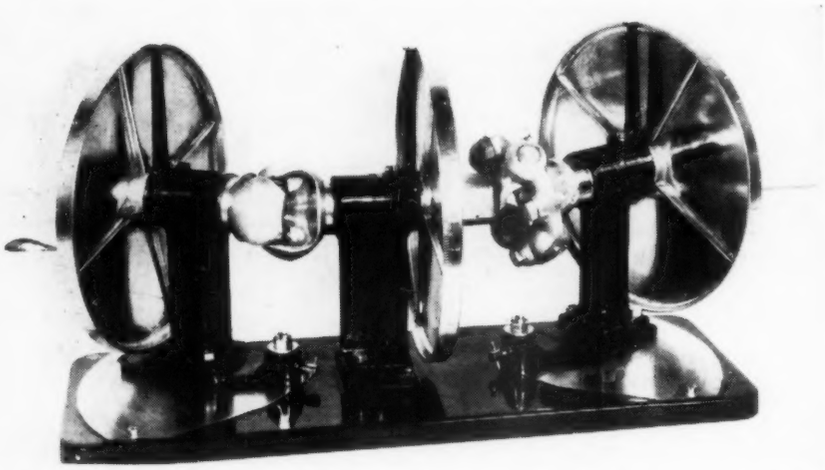


Fig. 6—A model with graduated dials for demonstrating uniform angular velocity feature of constant velocity and fluctuating velocity of the conventional joint.

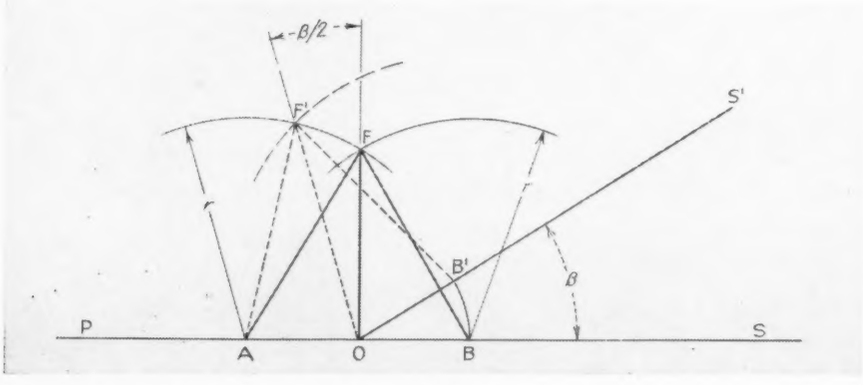


Fig. 7—Diagram used for geometric proof that power transmitting elements bisect the angle between driver and driven yokes in the Bendix-Weiss universal joint.

homokinetic joint is that all *PTE* are contained in the homokinetic plane and thus it follows that the *PTE* cannot be rigidly connected to, or be a part of either primary or secondary shafts *P* and *S*. Several designs use the principle of a double Cardan joint, being so closely coupled that the intermediary unit becomes the *PTE* following the previously stated laws of homokinetics. But by far the most compact designs have resulted from the revolutionary step of making the *PTE* load bearing steel balls, rolling freely in mating races of the yokes at the end of primary shaft *P* and secondary shaft *S* so that these balls are always arranged in the homokinetic plane.

A Bendix-Weiss homokinetic joint, Fig. 5, which has proven advantages in many applications for war and peacetime use where fluctuation-free transmission of power is important, is an extremely simple and efficient universal joint. Steel balls—usually four—operate as *PTE* in ball races in steel yokes. Without any forcible external means these balls arrange themselves at all times in the homokinetic plane.

This is achieved by machining curved mating races for each ball (*PTE*) into the two opposing yokes so that their center lines intersect, thus determining one definite location of this ball. The plane containing the centerline of each race passes through the axis of the shaft with which the yoke is often an integral part. The curvature of the races is generally either circular or a straight line, that is having a radius of curvature of infinity. By making the curvature of the two nonconcentric mating races identical in both yokes, but of opposite hand, the intersection point, determining the location of the engaging ball, will always bisect the angle between the *P* and *S* shafts. Thus all *PTE* (driving balls) are at all times in the homokinetic plane, which—as stated before—is the mathematical prerequisite of a homokinetic joint—more commonly referred to as "constant angular-velocity joint."

Geometrical Proof

Referring to Fig. 7, assume a race curvature on the primary yoke (left hand) to be represented by a circular arc with radius *r* and center of curvature *A* on the *P* axis. This race is mating with one on the secondary yoke (right hand) having the same characteristics, but of opposite hand and thus with its center of curvature *B* on the *S* axis. The intersection point of *P* and *S* is *O* and that of the two mating races *F* when *P* and *S* are in line.

Triangle *AOF* equals triangle *BOF* because both share the same side *OF;* side *AF* equals side *BF* equals *r,* the radius of curvature; and side *AO* equals side *OB* by construction. Consequently, angle *AOF* equals angle *BOF*. Since angle *AOF* plus angle *BOF* equals angle *AOB* equals 180 deg. or π, angle *AOF* equals angle *BOF* equals $\pi/2$ or 90 deg. so that point *F* lies in the homokinetic plane.

Now assume that *S* be displaced to *S'* by an angle β from *P*. The cen-

ter of the secondary race then moves to B' and the new intersecting point of two mating races becomes F'. Triangle AOF' equals triangle $B'OF'$ because both share the same side OF'; side AF' equals $B'F'$ equals r, the radius of curvature; and side OB' equals side OB equals side AO by construction. Therefore angle AOF' equals angle $B'OF'$; angle AOF' plus angle $B'OF'$ equals angle AOB' equals π minus β; and angle AOF' equals angle $B'OF'$ or $(\pi-\beta)/2$.

Thus point F' also lies in the homokinetic plane since it bisects the angle between primary and secondary axes P and S. Any displacement of angle β results in a shift by $\beta/2$ of the race intersection point which determines the location of the driving ball (PTE). Therefore, all PTE are at any instant in the homokinetic plane. The foregoing geometrical proof is accredited to John Varga Breuer of the Bendix Products Division.

This analysis is not only valid for circular race curvatures with centers on the P and S axes and for straight lines, but curvatures of any kind can be utilized and will produce a homokinetic joint so long as mating races do intersect, are nonconcentric, and are of opposite hand in the primary and secondary yokes.

Rolling ball contact between the two yokes, as in any ball bearing construction, reduces friction, promotes longer life and permits easy manual change of angularity even when power is being transmitted.

Another characteristic of this joint of interest to designing engineers is the ability to provide for end motion, when needed, within the joint itself. This is achieved by allowing the drive balls to roll back and forth in the races, doing away with an extra slip joint such as a sliding spline, and avoiding thrust and pull on bearings and other parts beyond the joint. A still further objection to sliding splines is that they must be kept well lubricated and at best backlash may develop, which is usually undesirable in mechanical drives.

Major items that determine the size and type of universal joint are:

Normal torque and maximum torque;

Normal speed and maximum speed;

Normal angle and maximum angle;

Extent of end motion, if any, required during operation;

Extent of additional end motion resulting from manufacturing variations;

Type of connection to joint, such as flange or splines;

If joint is not an integral part of design, is rotating or nonrotating type of housing desired; and

What are allowable clearances?

For helicopter and other aircraft designs it is especially important to have full information on all installation and operating conditions to provide for sufficient strength and durability for safety, but otherwise to hold weight to a minimum.

Case Histories In Patent Law

H. A. TOULMIN, JR.

[EDITOR'S NOTE: These explanations of the law in specific patent cases are not intended to supplant the services of a patent attorney, whose accurate advice is necessary to handle individual patent problems. The purpose of these explanations is to give a better general understanding of patent law.]

NO. 10

QUESTION: Can a patentee lose his patent in a patent infringement suit for reasons other than anticipation by prior uses, prior art or prior sales or publications?

ANSWER: "Yes." The Supreme Court of the United States has frequently pointed out, as in the recent case of Mercoid Corporation v. Mid-Continent Investment Company 320 U. S. 661, that the owner of a patent may not use it to secure a limited monopoly of an unpatented device employed in practicing the invention, even though the unpatented device is itself an integral part of the patented system.

In this case there was a patent for a domestic heating system and certain switches were supplied by the defendant only for use in connection with the system. This was the basis of a charge of contributory infringement. The switches were unpatented items but did form an integral part of the system covered by the patent.

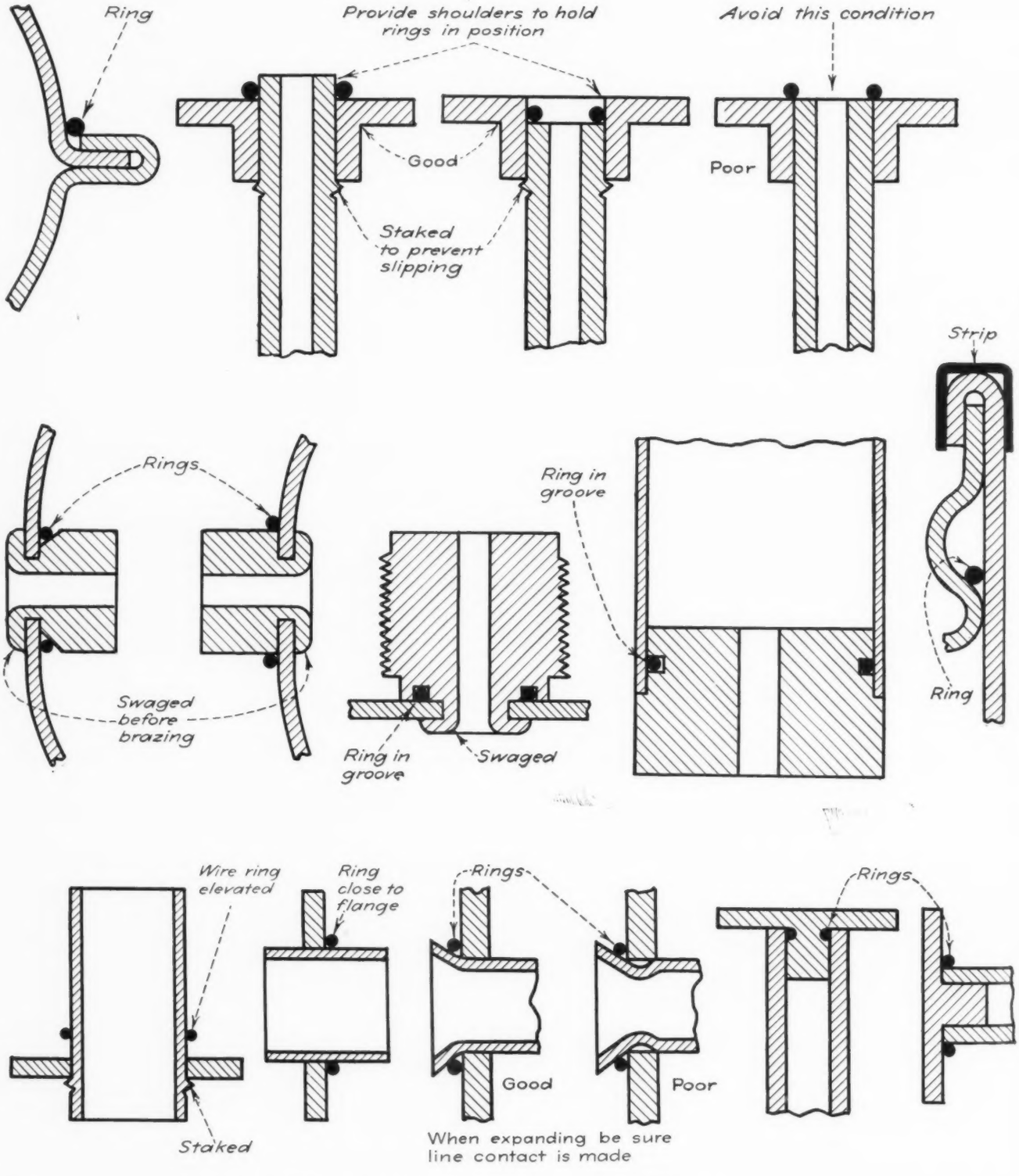
This attempted use of the patent to protect the unpatented switches from competition gave rise to a defense available to a contributory infringer. The court can withhold relief from a patentee who has misused his patent to secure a limited monopoly on unpatented materials like these switches.

In earlier Supreme Court cases such as the Motion Picture Co. v. Universal Film Co. 243, U. S. 502, it was held that the owner of a patent might not secure a limited monopoly on unpatented film used in the patented motion picture machines. Likewise in the case of Carbice Corp. v. American Patents Corp. 283 U. S. 27, the Supreme Court defeated the effort to control dry ice used in a combination patent. Likewise in Leitch Mfg. Co. v. Barber Co. 302 U. S. 458; Morton Salt Co. v. G. S. Suppiger Co. 314 U. S. 488; and B. B. Chemical Co. v. Ellis 314 U. S. 495; the patentee lost his patent by showing the patent he was using "as the effective means of restraining competition with its sale of an unpatented article."

That is because the public interest is dominant in the patent system. It is against the public interest to allow a patentee to use it in such a way as to acquire monopoly which is not plainly within the terms of the patent granted. The patent is limited to the invention which defines it and any attempt to control other than the complete combination of the claim where the other thing controlled is an unpatented element will result in disaster for the owner of the patent.

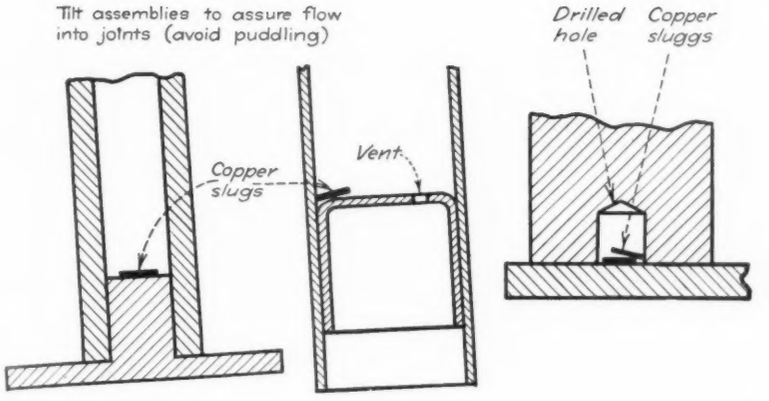
METHODS FOR PLACING BRAZING

Location and forms of copper used in brazing. Suggested positions for vents to prevent deformation of the final product. Two examples of joining dissimilar materials together by brazing. Sketches courtesy the General Electric Company.

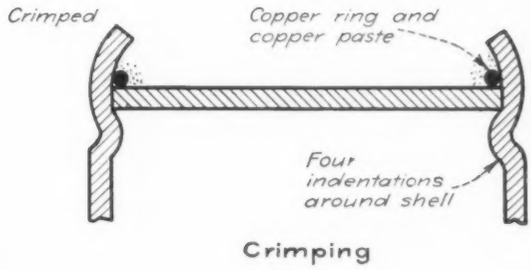


Placement of Copper Rings

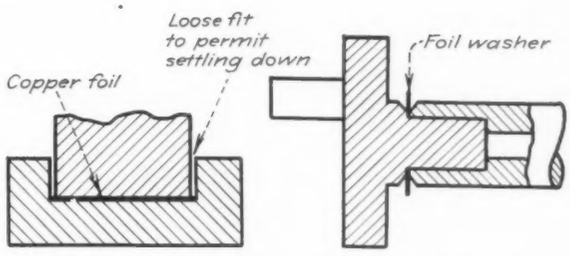
MATERIALS, AND VENT LOCATIONS



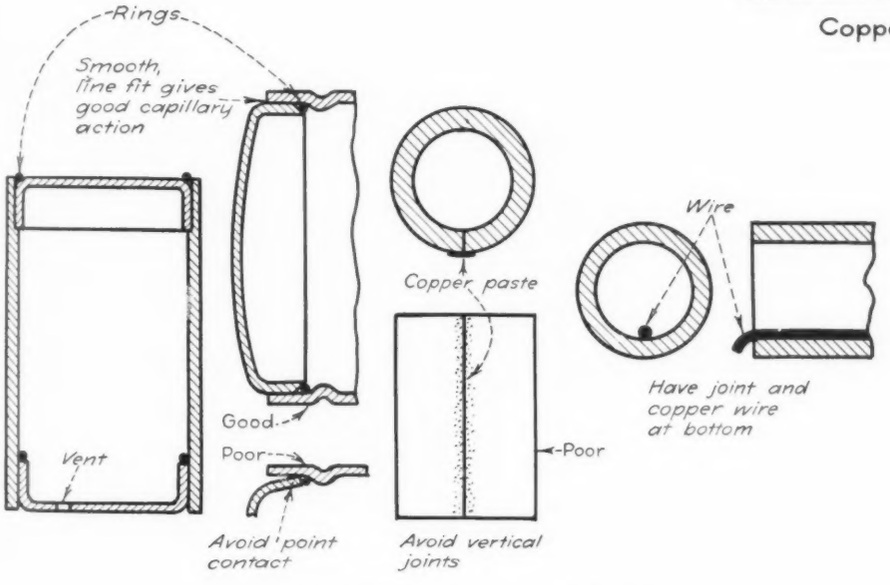
Copper Slug Positions



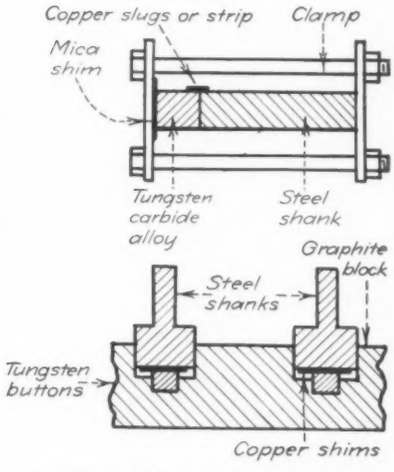
Crimping



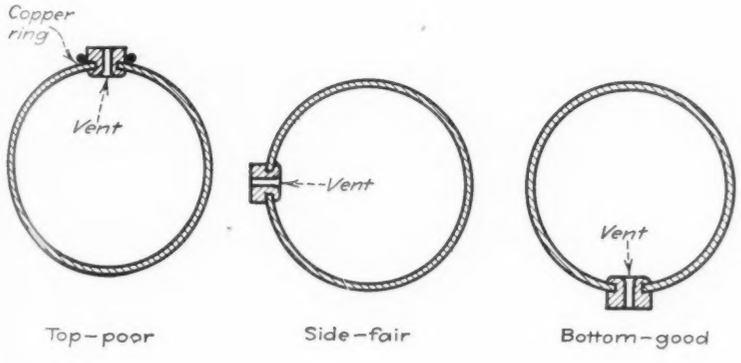
Copper Foil Used Inside Joints

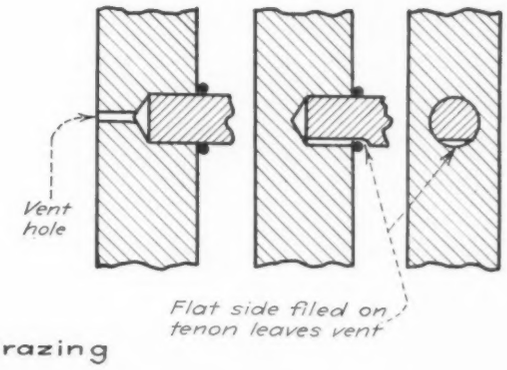


Placement of Various Forms of Copper



Joining Dissimilar Materials





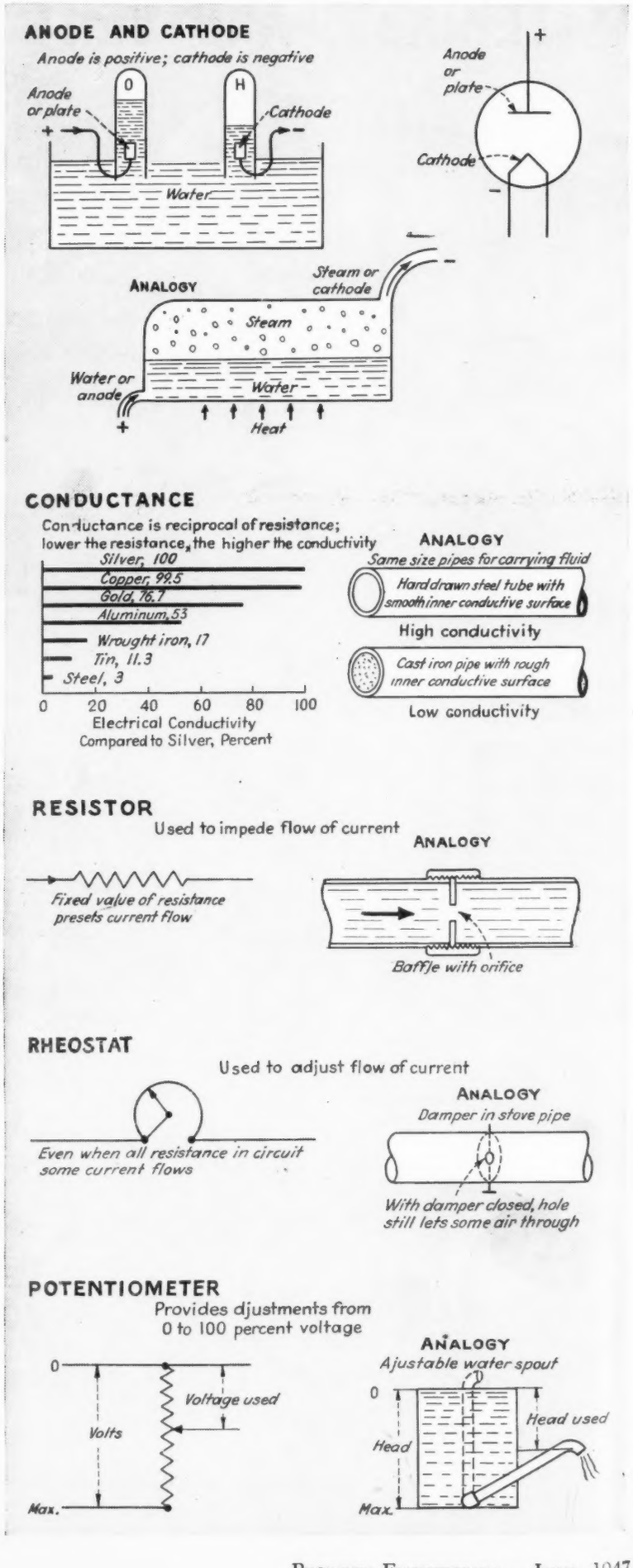
Proper Venting During Brazing

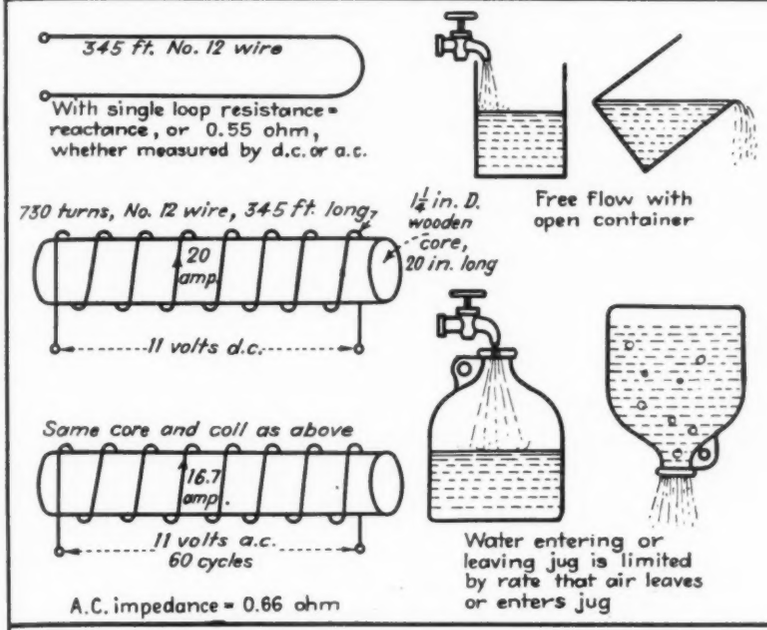
SOME SIMPLE ELECTRICAL ANALOGIES

* * *

When explaining the action of electrical principles, engineers often find that analogies are the simplest and quickest means to make themselves understood. Here is a group of handy analogies that Record Rogers of the Industrial Engineering Department of General Electric has assembled. Some of the analogies are fairly exact, while others merely give a general physical concept on which to build a more precise engineering explanation.

* * *




345 ft. No. 12 wire
With single loop resistance = reactance, or 0.55 ohm, whether measured by d.c. or a.c.
730 turns, No. 12 wire, 345 ft. long
1¼ in. D. wooden core, 20 in. long
20 amp.
11 volts d.c.
Same core and coil as above
16.7 amp.
11 volts a.c. 60 cycles
A.C. impedance = 0.66 ohm
Free Flow with open container
Water entering or leaving jug is limited by rate that air leaves or enters jug

REACTANCE

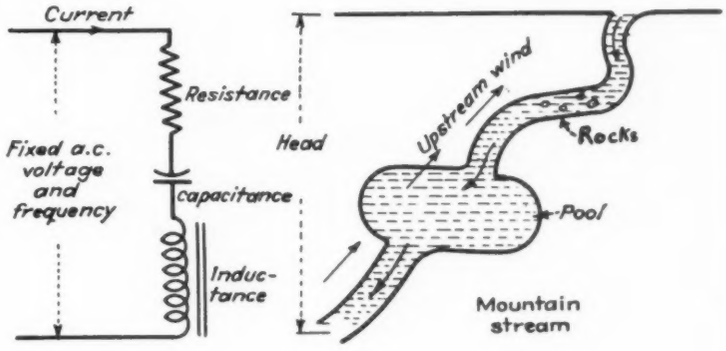
When the resistance of a circuit carrying alternating current is greater than its measured d.c. resistance, the circuit possesses reactance; that is, there is some condition present that reacts against the voltage. Reactance is caused mostly by the magnetic field that a current always sets up about itself, which creates a counter or opposing voltage that is proportional to the frequency.

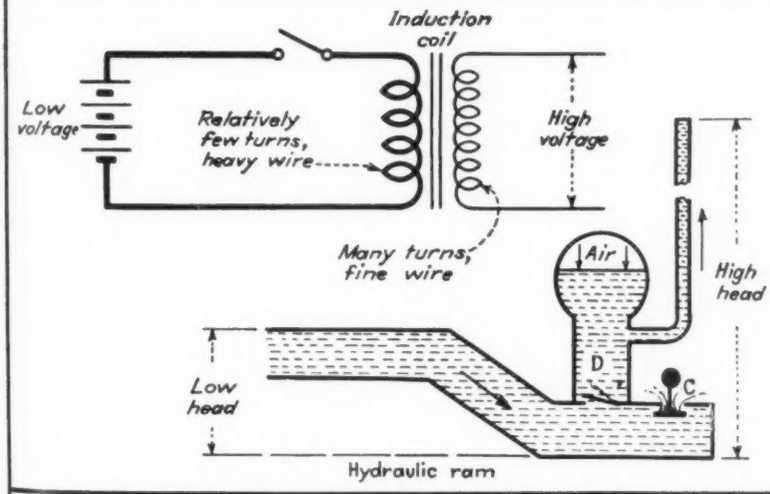
ANALOGY. A hydraulic analogy of this effect is filling a jug with water and then emptying it. When water is poured in through the small opening, the rate at which it will enter is limited by the rate at which the air can escape. When the jug is full and turned upside down and emptied, the water can leave only as fast as the air can enter. In either filling and emptying, the air is analogous to the effect of reactance in an electrical circuit.

IMPEDANCE

Impedance, or "apparent resistance", is a property of an a.c. circuit that depends upon resistance, capacitance, inductance and frequency. For a given voltage and frequency the other combined factors determine the amount of current that will flow in the circuit.

ANALOGY. Impedance is analogous to a mountain stream. The stream has a pressure head or voltage, the curves in the stream, the boulders, the fallen trees and brush are essentially resistance, the pools act like capacitors, and the upstream wind gives an inductive effect, all of which limit the rate of flow of current. The effect of these things together would be the impedance.

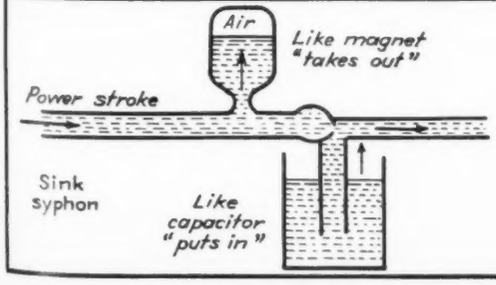




INDUCTIVE KICK

If voltage is applied to a pure inductance and then theoretically interrupted in zero time, the current is reduced to zero instantly and the voltage is infinity. This principle has many uses; for example, by powering an induction coil from a 6 volt battery, automobiles obtain approximately 8,000 volts for firing spark plugs.

ANALOGY. Action of an hydraulic ram is analogous to that of an induction coil. Water flows down a smooth pipe with little resistance. The height from which the water comes is the operating pressure head; or electrically, the supply voltage. The rate of flow is the current. Water first escapes through valve *C* that opens downward, but as the speed of the water increases, it catches the valve in its rush and shuts it. This sudden stoppage of the stream forces the water up through *D* because of the water's stored energy. By this surge action, a hydraulic ram can raise fluid many times higher than its operating head.



CAPACITORS

Capacitors are devices for increasing the capacitance of an electric circuit, principally to counteract inductance. However, they have many other uses, such as for accurate timing in electronic application, or for slow storing of energy for sudden release. An inductive device, such as an a.c. magnet winding, absorbs energy in each cycle when the power is rising and gives it back when the power is falling. A capacitor does directly the opposite.

ANALOGY. When connected in ordinary circuit, a capacitor plays "put-and-take" while the inductive devices play "take-and-put." This is similar to the action of a sink syphon.

(*Continued on next page*)

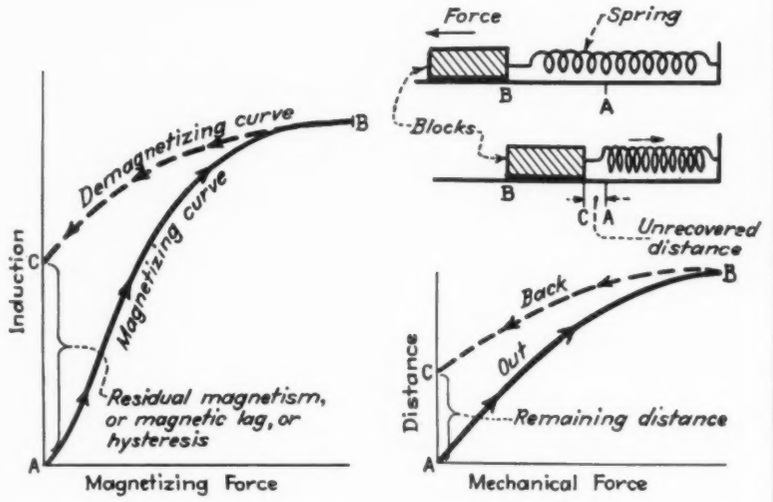
HYSTERESIS AND RESIDUAL MAGNETISM

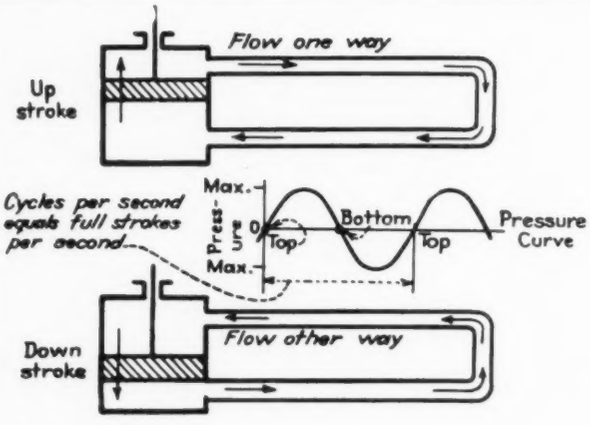
A magnet does not cease to be a magnet as rapidly as it becomes one. This time lag is called hysteresis and is caused by residual magnetism.

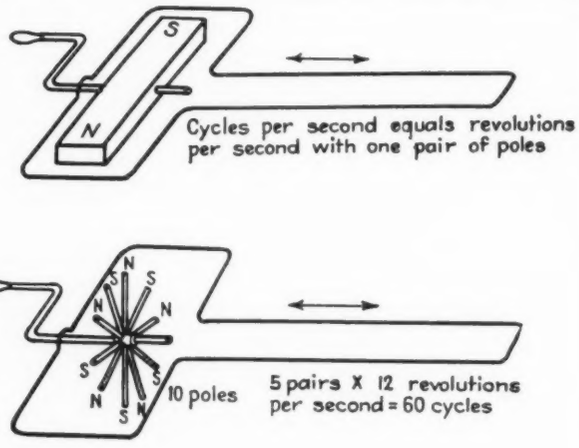
On the curve starting at *A* and going to *B*, the rate of magnetization is shown. Going from *B* to *C* shows the rate of demagnetization. The distance from *C* to *A* is the residual magnetism. The time taken for the return from *B* to *C* to *A* is the hysteresis for the particular metal. When an electromagnet is energized and de-energized rapidly heat is generated and this energy loss is an important design factor.

ANALOGY. Once a magnetic material is magnetized it retains some residual magnetism indefinitely unless removed by a counter-force. Consider the action of a block and spring. When the block is released the spring pulls it back toward the starting point, but the spring will not pull it all the way back. This remaining distance corresponds to the residual magnetism in a magnet.

The reversed magnetizing force that is just enough to reduce the residual magnetism to zero is the coercive force. In the block and spring, the amount of force necessary to return the block to the starting point is the coercive force.






S
N
Cycles per second equals revolutions per second with one pair of poles
10 poles
5 pairs X 12 revolutions per second = 60 cycles

CYCLES

In an alternating-current circuit, the cyclic values of volts and amperes rise from zero to a maximum value and back to zero, then to a maximum value in the opposite direction and back to zero. An important factor in an alternating electric system is frequency, or the number of times per second these round trips are made.

ANALOGY. Actually electricity moves back and forth in the circuit just as water does in the pump actions shown. The generator delivers power to the circuit just as the piston rod delivers power in a reciprocating manner. One complete round trip of a piston is one cycle.

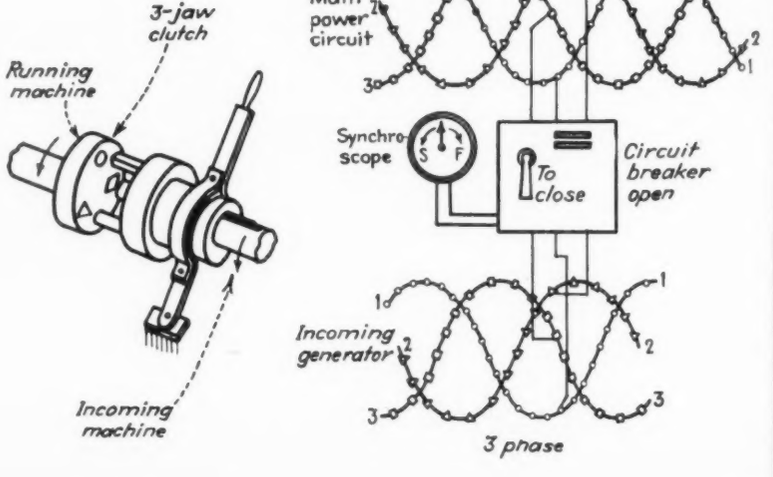
The frequency of a generator is determined by the speed and the number of pairs of poles. If a generator runs at a speed of 720 r.p.m., or 12 rev. per sec., and has 10 poles or five pairs of holes on the rotor as shown in the sketch, the product of 12 rev. per sec. and five pairs of poles is 60. Thus the generator supplies 60-cycle power, which is the most common.

PHASING-IN

When coupling two 3 phase a.c. generators or circuits together, the sources of power must be phased-in or synchronized with each other before their circuits are joined.

ANALOGY. Consider a three-jaw clutch that is to connect an incoming machine to a running machine. If the jaws were all alike the job would be comparatively simple. But with a round, a square, and a triangular jaw and corresponding holes to match up, both speeds must be the same and the projections must be opposite the corresponding holes before the clutch can be engaged. Closing the clutch in under any other condition would make a mess of things, just as it would if two circuits were connected out-of-phase.

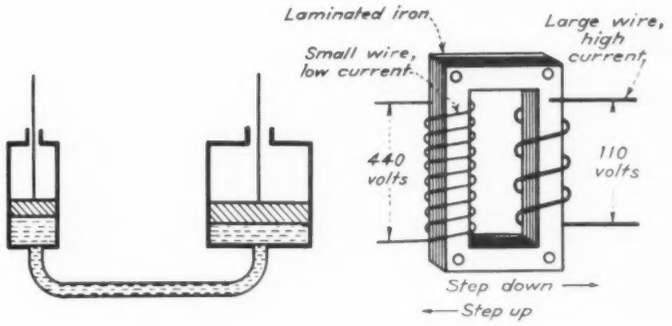
The electrical diagram shows the three phases of the running side marked with circles, squares, and triangles designating phases. Below the circuit breaker are shown the corresponding phases of the incoming circuit. In the sketch the two circuits are exactly in phase; the synchroscope is standing still at "12 o'clock," showing conditions are right to phase-in.

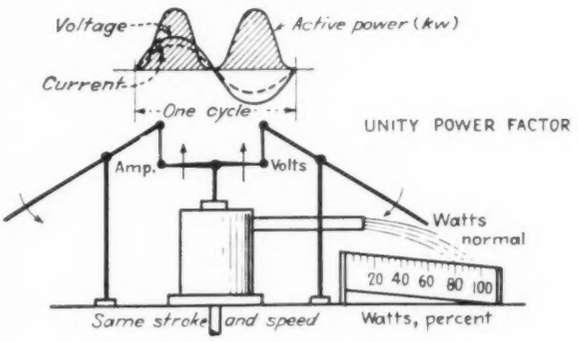


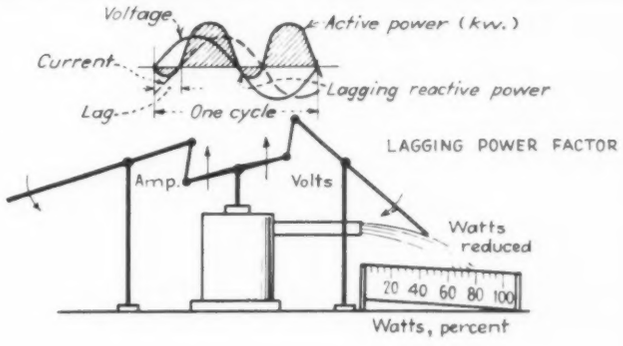
TRANSFORMER ACTION

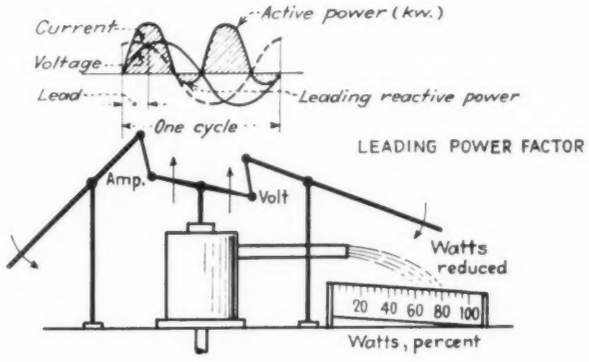
Transformers change voltage levels or current levels and they are important in most a.c. circuits. The ratio of voltage change is the same as the ratio of turns in the incoming coil to the turns in the outgoing coil. Amperes vary the opposite way but in the same ratio. The same transformer may step-up or step-down, according to which coils are connected to the incoming and outgoing circuits. Frequency remains unchanged.

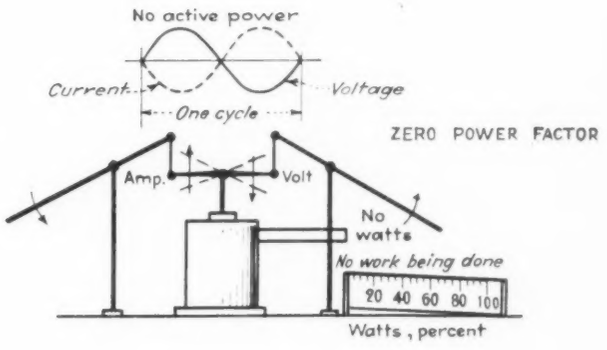
ANALOGY. This transformer action is similar to the hydraulic system shown. The small piston represents the winding of small wire and the large portion the winding of large wire. When the small piston is moved 4 in. the large piston moves one inch, or a 4 to 1 ratio of motion.











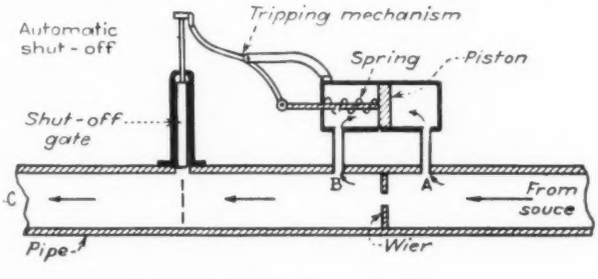
POWER-FACTOR

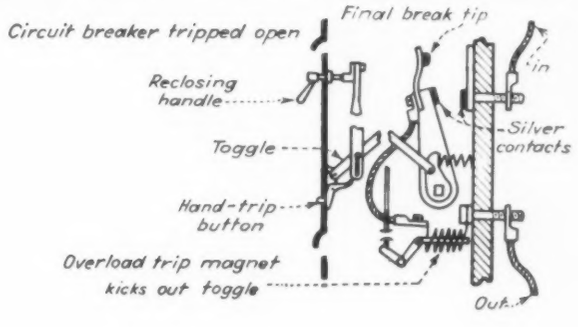
In a non-inductive circuit of only resistance, the power is equal to the product of volts times amperes, whether the current is alternating or direct. The current is always in phase with the voltage. But in a circuit containing both resistance and inductance, the a.c. current lags behind the voltage and so the apparent power, or volt-amperes, is greater than the true power. A similar condition occurs when the circuit has resistance and capacitance except that the a.c. current leads the voltage.

Power factor is the ratio of true power (watts) to apparent power (volt-amp.). Power factor is classified as unity, leading, and lagging; the last two ranging anywhere from unity to zero.

ANALOGY. In the two-handled pump analogy, when both are working together with the same stroke and speed, the water output representing watts is normal. When the pump marked Amp. lags behind Volts, the watts are reduced by a lagging power-factor. When Amp. get ahead of Volts, watts are again reduced as a result of a leading power-factor.

When the point is reached where Volts is at one end of the stroke when Amp. is at the other, and vice versa, there is no watts output and the condition is called zero power-factor. That part of the current that is not working is called wattless current, but this loafing current heats up conductors as much as the working current does, just as the linkage of the pump continues to work although no water is pumped.





CIRCUIT BREAKERS

A circuit breaker opens and closes an electric circuit. It is operated manually, electrically, or pneumatically, primarily used as a protective device to open the circuit when current exceeds the predetermined safe value. This is done by an overcurrent trip coil and mechanism.

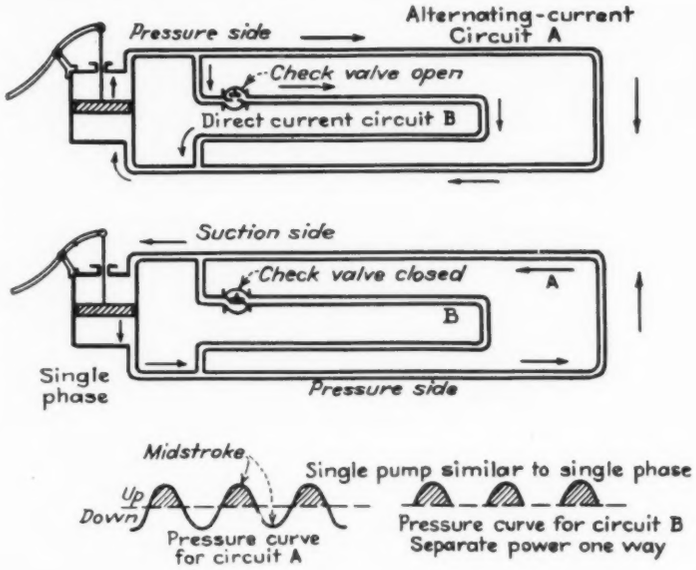
ANALOGY. Hydraulic shut off gate operates when the pipe breaks at *C*, letting an excessive current of water flow. Increased flow creates lower pressure at *B* than at *A*, piston trips gate, which falls and shuts off water.

(Continued on next page)

RECTIFIERS

In rectifying alternating-current to direct-current, the galena crystal, copper-oxide plates, selenium plates, vacuum tubes, and mercury-arc rectifiers, all depend upon check-valve action. Like a check valve in a water pipe, they pass current more freely one direction than in the other. When the current reverses, they clap shut until the next forward current comes along.

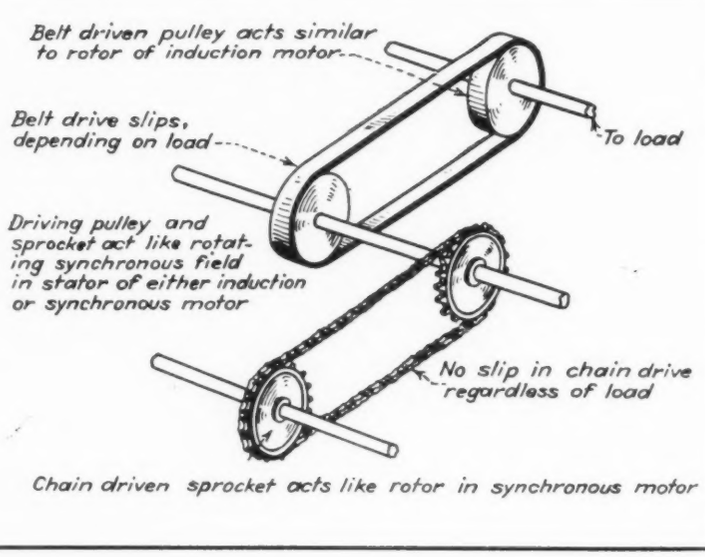
ANALOGY. In the alternating-current water system pressure is first on the upper side and suction on the lower, and then conditions are reversed. In the first diagram, water passes down the pipe *B* through the open check valve. During the reverse cycle the check valve shuts and there is no flow in pipe *B*. This cycle is repeated over and over. What water current goes through *B* is always in one direction. The pressure curve in pipe *A* is a sine wave. That for *B* is a series of pulses separated by equal blank spaces. This is unidirectional but pulsating current. Rectifier circuits smooth out these pulses.

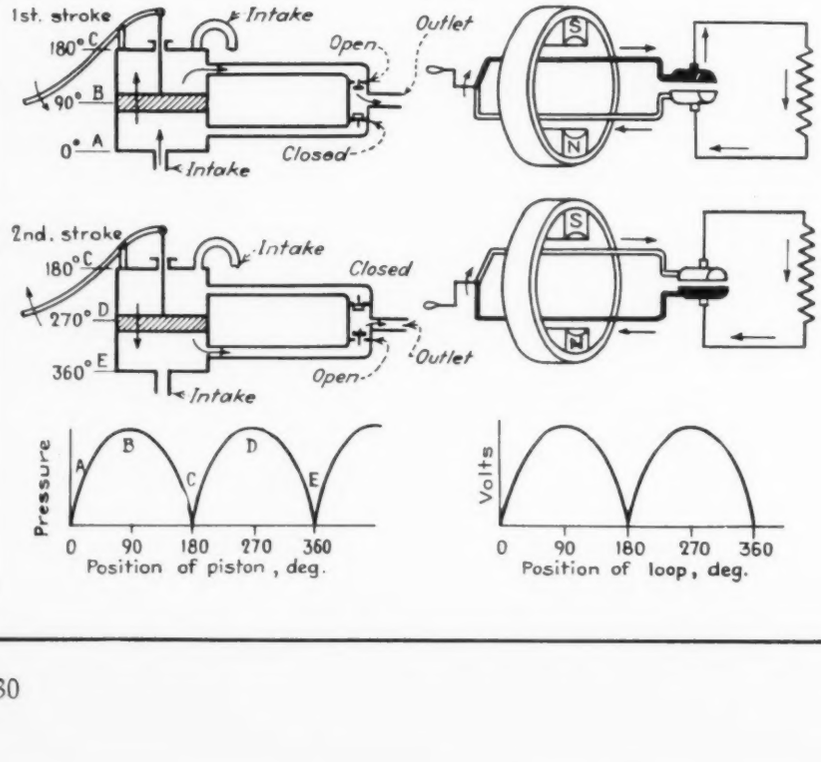


SYNCHRONOUS MOTORS

Synchronous motors are used on a.c. circuits. Speeds are exact; there is no "slip" at any load. For a given frequency, the speed is not adjustable. Cycles per minute divided by pairs of poles determines speed in r.p.m. For 60 cycles, speeds include 3,600, 1,800, 1,200 and 900 r.p.m. A synchronous motor starts and reaches about 97 percent of speed as an induction motor and when the field circuit is closed it pulls into step or synchronism.

ANALOGY. The belt and chain drive are similar to the operating principles of an induction and synchronous motor, respectively. With a heavy load, the belt driven pulley slips and drives the load at reduced speed just as does the rotor of an induction motor. But like the chain drive, the rotor of the synchronous motor keeps the load turning at constant speed up to its rated capacity.


Belt driven pulley acts similar to rotor of induction motor
Belt drive slips, depending on load
To load
Driving pulley and sprocket act like rotating synchronous field in stator of either induction or synchronous motor
No slip in chain drive regardless of load
Chain driven sprocket acts like rotor in synchronous motor



COMMUTATION

In d.c. machines, although the direction of current flow inside the armature coils reverse as they pass under opposite poles, a commutator keeps the flow of current in the external circuit always in the same direction.

The commutator consists of wedge shaped insulated bars around the shaft, with a separate bar for each coil. Split "cans" on the ends of a single coil and brushes form a commutator as shown. In the first position, current is flowing from the black wire to the black commutator bar. In the second position 180 deg. later, current in the black wire is flowing in the opposite direction. The current reverses in the wire, but the commutator bars changed places just in time to fool it.

ANALOGY. Commutation is analogous to a double-action pump with two inlets and one outlet. The current in the pump reverses with the piston but the outlet current is always in the same direction.

ENGINEERING ABSTRACTS

FOREIGN AND DOMESTIC ARTICLES

Designing Small Ceramic Parts

Foreign Abstract, condensed from an article by J. Wallich in V. D. I. Zeitschrift, Bd 88 Nr. ½, 8 January 1944.

The design of ceramic parts differs from that of metal parts because the shape of ceramic parts is substantially produced by a baking process. Ceramic parts are nonelastic and must not be subjected to bending loads or high stress concentrations. If such loads cannot be avoided, elastic supports become necessary.

Ceramic parts shrink during the baking process, which usually takes place at about 2,550 deg. F. Shrinkage varies with the type of ceramic material to some extent and must be taken into consideration in the design. It is, for example, not advisable to use round holes in plates, gibs and similar parts but better to provide oval shaped holes so as to compensate for shrinkage of parts that must be assembled with other parts.

Finishing of ceramic parts is possible only by wet grinding at high speeds. This technique has been improved by the ceramic industry in recent years to such an extent that an accuracy of ±0.0004 in. can be obtained by simple plane or cylindrical grinding. In plane parallel grinding an accuracy of ±0.0012 in. per 4 in. length is possible. Shafts and axles can be centerless ground to limits of 0.0012 in. per 4 in. for pieces not exceeding a total length of 28 in., and to limits of 0.0016 in. per 4 in. for longer pieces. The smallest hole that can be ground to a tolerance of ±0.0012 in. is about 0.16 in. in diameter.

Threads can be cut in ceramic parts on lathes, but it is advisable to cut V-threads only. Round threads should be avoided because of the difficulties in dressing the wheels. The foregoing tolerances correspond to good average practice. They can be reduced whenever closer tolerances are required and the extra expense is justified. The following tolerances can be obtained on ceramic parts without a finishing process:

± 3 percent in turning or casting but not less than ±0.0120 in.

± 2 percent in wet pressing but not less than 0.0080 to 0.0120 in.

± 1½ to 2 percent in dry pressing but not less than 0.0060 to 0.0080 in.

Surfaces that must be ground should be kept small in order to reduce grinding time. Fig. 2 shows examples. Grinding wheels dull rapidly on ceramic material and lose profile. A blending radius of at least 0.024 in. is therefore required for grinding steps or similar profiles.

Form Presses. Wet or dry presses equipped with steel dies are mostly used for mass production of small and medium sized ceramic articles. In the wet pressing process, the ceramic material—with water and oil added—is piled up in the lower die and compressed by the upper half of the die until the cavity is completely filled. Excess material can escape through holes in the die.

In the dry pressing method water is not added to the material. The amount of ceramic material must be exactly predetermined because it cannot escape from the die. The downward moving portion of the die compresses the material until it fills about half the cavity. Dry pressed parts are therefore more compressed than wet pressed parts and of higher quality in mechanical and electrical properties.

The dry press method, on the other hand, is disadvantageous in that it is not possible to produce thin walled parts or parts with undercuts. The wet method gives excellent results on such parts and allows complex shapes. Arti-

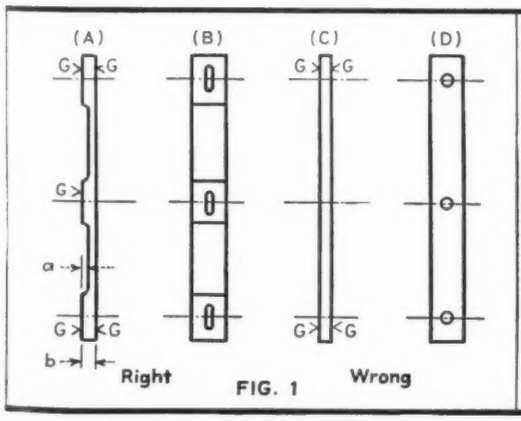


FIG. 1

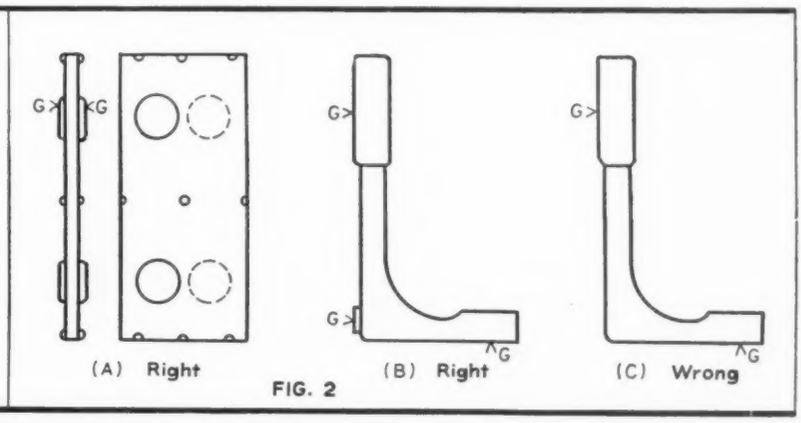


FIG. 2

Fig. 1—(A) Ears on gibs are less costly to grind than (C) gibs of uniform thickness; (B) Oval holes allow for variations in shrinkage whereas round holes (D) cause difficulties in assembly. Fig. 2—Parts designed with ears (A) to support the ends during grinding and baking and (B) to facilitate clamping during grinding are more economical to manufacture.

cles produced by the dry press process are generally more accurate than those coming from wet presses because they do not shrink before baking. Figs. 3, 4 and 5 give a variety of examples.

EXTRUSIONS. Tubes, bars and other pieces of appreciable length are produced by extrusion of ceramic raw material, Fig. 6. The greatest length is about 28 in. for tubes up to 10 in. in diameter, while bars up to 3 in. in diameter can have a length up to 60 in. Also, small tubes can be produced in this way, for example, tubes with 0.0080 in. wall thickness and 0.160 in. in diameter.

CASTING. Ceramic parts can be cast with and without the use of cores. In the coreless casting process, water, sodium carbonate and potassium silicate are added to the ceramic raw material, which is then cast into the mold made of plaster of paris. Because of the porosity of the walls of the mold, the water is rapidly absorbed, leaving a coating of ceramic material on the wall. When this coating has reached a desired thickness, the rest of the liquid ceramic material is removed. As the coating dries, it shrinks and the part can easily be removed after a few hours. Complex parts can be produced in this manner with uniform wall thickness.

Bodies with varying wall thickness and such parts where the inner contour is not the same as the outer contour are cast with cores made of plaster of paris. In these applications the same rules apply regarding design as for casting metal parts.

TURNING. Ceramic parts can be turned when pressed into a mold of plaster of paris, using the principle of the potters wheel. The ceramic material must be "leather hard" and can be turned with ordinary lathe tools. Even fine threads can be cut with high accuracy. The rules for turning ceramic parts are the same as those for metal parts. For instance sharp corners must be avoided and a radius be provided where one diameter blends into another.

SINTERING. Large ceramic articles can be produced by assembling small, unbaked units. The assembled part is coated with liquid material and baked in an oven, where a sintering process takes place uniting all component parts solidly into a single body.

ASSEMBLY-GLAZING. Sintering must not be confused with "assembly glazing" which is a different method of manufacturing large ceramic parts by joining smaller parts into a single body. In assembly glazing the component parts are baked and, if necessary, ground before assembly, while in the method of sintering the component parts are in a "soft" state when assembled, and only dried.

Assembly glazing takes its name from the fact that a glazing liquid is used as a binder for producing a single body. It is possible to obtain an accuracy of a few thousands of an

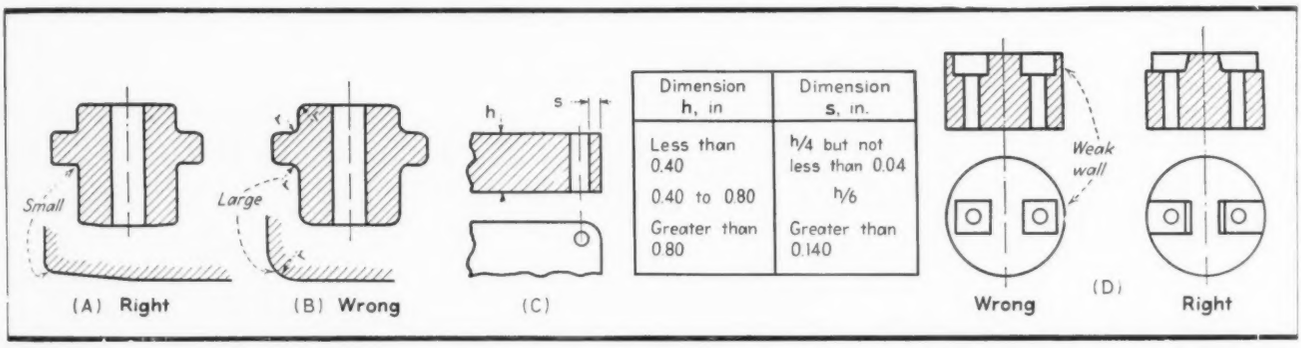


Fig. 3—(A) Lead angle instead of a radius (B) reduces tool cost. Small radii are desirable on dry pressed parts. (C) Recommended dimensions for holes. (D) Open, square-head design avoids weak wall during manufacture and afterward.

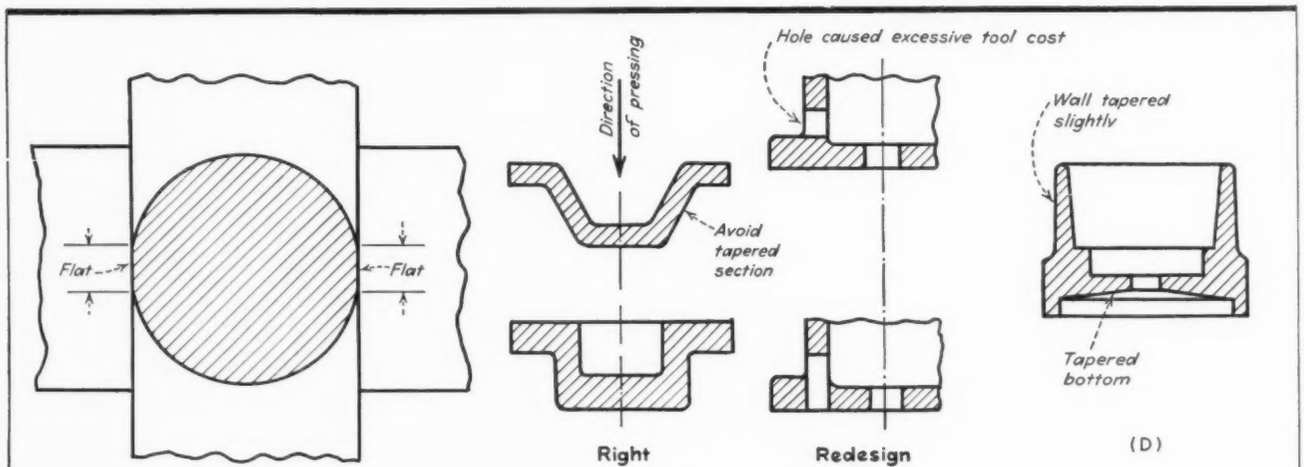


Fig. 4—(A) Flats at the sides of circular horizontal sections avoid sharp corners on the dies. (B) Surfaces perpendicular to the direction of pressing rather than tapered surfaces are the easier to compact uniformly. (C) Side hole redesigned to simplify tooling and speed up production. (D) Rounded corners and tapered walls facilitate pressing and avoid warping.

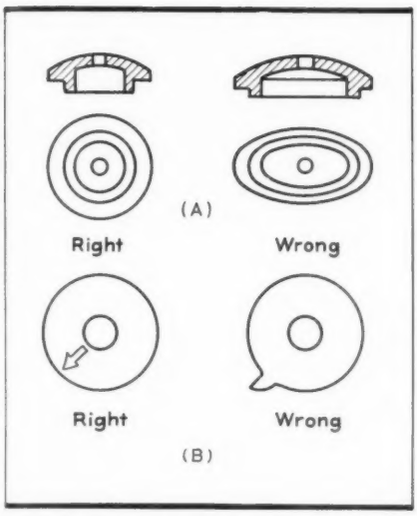


Fig. 5—A round body without undercut requires less expensive tools than oval shape and undercut. (B) Raised arrow on round piece is simpler and cheaper to make, and less likely to break off.

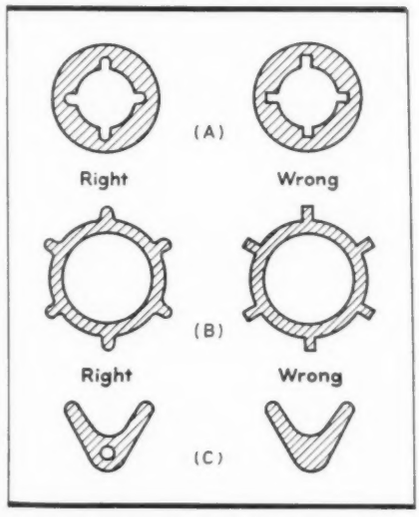


Fig. 6—Design extruded shapes (A) to have rounded corners; (B) to have uniform wall thickness. (C) The hole in section results in good mass distribution and approximately a uniform wall.

inch by assembly glazing, while only ± 2 percent accuracy would be possible if the complete part were made in a mold.

Assembly glazing can also be used where parts of slightly different coefficients of expansion are to be united into a single body. The temperature at which the binder liquifies lies between 1,550 and 1,650 deg. F. which is lower than the baking temperature of the individual part. The parts are therefore not affected by the glazing process; this explains the high accuracy of assembly glazing.

Metals can be joined with ceramic parts either mechanically or by electrothermical processes or by silver coating. The last method, which eliminates any air gap between the metal and the ceramic body, permits soldering which is particularly desirable for high frequency instruments. Copper plating after silver coating of ceramic parts allows manufacturing of coils and condensers with unsurpassed reliability.

Surface Hardening of Aluminum

From "Surface Hardening of Aluminum and its Alloys" by K. G. Robinson and B. W. Mott in *Metallurgia* (England), February 1947, page 201.

An account is given of some experiments made to investigate the possibility of hardening the surface of aluminum and the alloys of aluminum. Of the methods investigated, copper, which was known to form with aluminum intermetallic compounds of high hardness, appeared to be a suitable hardening agent. The process developed for diffusing the copper to the aluminum alloy was as follows:

Duralumin samples in 16 S.W.G. sheet measuring 1 x 0.5 in. were degreased in trichloroethylene and pickled in a solution containing 5 g. per liter of sodium cyanide until no further water-break was observed. Following cleaning the specimens were anodized for 10 min. in an electrolyte containing 30 g. oxalic acid per liter at 50 v. a-c with a bath temperature of from 16 to 21 deg. C. Similar samples were disposed in the electrolyte so that each served as a counter-electrode for those next to it.

After the anodic treatment, the samples were rinsed in cold water and modified in a solution containing 60 g. of sodium cyanide per liter. As experience was gained, completion of modification could be judged fairly accurately. Once the end-point for a set of conditions had been determined, the process was carried out on a time basis. A surface prepared in this manner was then ready for the plating operation, which was carried out at a current density of 20 amp. per sq. ft. at room temperature in a bath containing 200 g. of hydrated copper sulphate and 20 milliliters of sulphuric acid per liter.

A thin coating of nickel, 0.00025 in. in thickness was electroplated onto some of the samples, prior to the usual coating of copper, 0.0005 in. thick, in an attempt to obtain better adhesion between the coating and the aluminum. The reasoning was that the nickel to aluminum interface would be better than the copper to aluminum and that the diffusion of the copper would take place through the nickel, although the composition after diffusion would not be known. Further, in an attempt to reduce oxidation of the copper during heating without the use of a controlled atmosphere, some samples were given a flash of nickel over the copper.

Diffusion of the copper was obtained by heating at 500 deg. C., as this temperature was considered to be the optimum safe temperature of heat-treatment for duralumin. Other aluminum alloys can be heat-treated at higher temperatures without risk of overheating the base metal, with a consequent increase in the rate of diffusion of the copper and a marked decrease in the time required to produce a homogeneous layer of $CuAl_2$. It is considered that if the coating were sound and homogeneous, the corrosion resistance would be good, but in the event of an unsound coating or rupture during service, then the corrosion resistance would be considerably reduced.

At a temperature of 500 deg. C., copper oxidizes fairly rapidly so that the loss of copper as oxide was appreciable in an uncontrolled atmosphere. This meant that the copper available for diffusion was a variable quantity, depending upon the degree of oxidation during heat-treatment. Several controlled atmospheres were tried. An atmosphere of oxygen-free nitrogen was found suitable. The duralumin samples were quenched in cold water from the diffusion temperature of 500 deg. C. and aged for two days at room temperature to facilitate polishing.

A number of samples were examined in the "as plated" condition to check the uniformity and adhesion of the coating. The modified anodic film was always present as a dark interface and appeared to be continuous and uniform. In general the adhesion of the copper was good for the samples plated with copper alone as well as for those plated with both nickel and copper.

Examination showed that diffusion had taken place with the formation of two clearly defined layers of different composition. The outer layer adjacent to the remains of the copper plating was identified as the CuAl phase; the inner band adjacent to the parent metal

was identified as $CuAl_2$. The boundary between the two layers was sharply defined, the two layers being distinguishable by differences in color and polishing characteristics. The copper-rich phase appeared to be the more brittle. The aluminum-rich phase seemed to be reasonably ductile, no difficulty being experienced in preserving it during polishing.

Hardness of the coating was determined with a Tukon Micro-hardness tester, using both a standard Vickers diamond indentor and a Knoop diamond indentor. The Knoop indentor produces a diamond-shaped (rhomb) indentation having long and short diagonals of approximately a 7 to 1 ratio. The Knoop indentor was used for the majority of the tests because the indentations from light loads were easier to read than those of Vickers indentor.

The Knoop indentor and a load of 50 g. indicated a Knoop hardness of the copper-rich phase (CuAl) of approximately 550 and of the aluminum-rich phase ($CuAl_2$) of approximately 420. On the Vickers Hardness Scale the corresponding hardnesses of the two phases were 580 and 390 V.P.N., respectively, which are in fairly close agreement with the Knoop hardness values thereby verifying test methods.

A.C. Hoist Lowering System

From "A Thrustor Lowering Control System for A.C. Crane Hoists" by E. L. Schwarz-Kast, Illinois Institute of Technology, presented A.I.E.E. Jan. 1947, New York, N. Y.

Because of crane load characteristics, the speed control for hoists is more exacting than that for general industrial drives. As compared to the usual changes in load from zero to full load, in a crane hoist the motor load may change from zero to full load during hoisting and from +10 percent for lowering the empty hook to —64 percent for lowering the full load. Also because of the danger of dropping a load abruptly, safety in crane control is essential.

As shown in Fig. 1, the five ideal characteristics for hoist control are: (1) To hoist all normal loads at slow, medium and high speed; (2) to hoist very light loads and empty hook at slow and high speeds; (3) to lower all loads at a slow, medium and high speed; (4) to provide accurate positioning of all loads, overhauling or non-overhauling (so-called inching); and (5) to avoid excessive currents to reduce heating of motors and apparatus.

Since a.c. is universally available in industry, different a.c. systems are striving to meet the need for cranes with higher operating speeds and greater accuracy of positioning.

Commonly used systems for a.c. lowering include:

1. Oversynchronous lowering with squirrel cage and wound-rotor motors.
2. Multi-speed motors, squirrel cage, wound-rotor or combined wound-rotor, squirrel cage motors.
3. Mechanical load brake, wound-rotor motor and rotor resistors.
4. A.c. commutator motor with shunt characteristic and brush shifting.
5. Dynamic braking with wound-rotor or squirrel cage motors and d.c. applied to the stator windings.
6. Counter-torque (plugging) with wound-rotor motor in combination with oversynchronous lowering.
7. Wound-rotor motor, rotor resistances and unbalanced voltage, using an autotransformer with multiple taps in one phase.
8. Variable unbalanced voltage. The unbalance is obtained by a saturable reactor in one motor lead. This reactor is saturated automatically in proportion to the motor speed. The rotor voltage is used as a simple and reliable speed indicating means that varies inversely proportional to the speed. The rotor voltage furnishes, through an amplifier and rectifier, a d.c. output inversely proportional to the speed that is used in the reactor for obtaining the necessary unbalance of the primary voltage.
9. Variable voltage control, or Ward Leonard system.
10. Thrustor lowering, relatively new electro-hydraulic system.

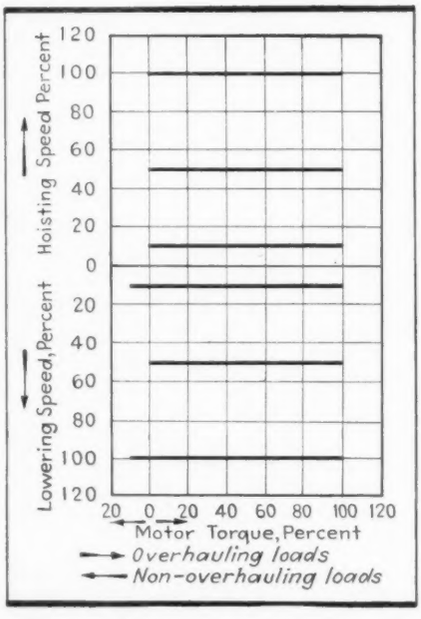


Fig. 1—Ideal speed-load diagram for correct performance of a crane hoist.

The main properties of the various systems are shown in Table I. The approximate initial cost of the electrical hoist equipment is compared with the cost of No. 1 or oversynchronous lowering, which is the lowest and is assumed as 100 percent. The last column shows some disadvantages. This table shows very clearly that there is a definite demand for an a.c. lowering system that is simple, safe, has low current consumption, assures reasonable speed reduction and easy inching, and is inexpensive.

The Thrustor type system, first applied about six years ago, has many of the desirable characteristics for a.c. lowering control. The system uses a standard wound-rotor crane duty motor for the drive and a Thrustor operated stop-brake to develop the necessary motor load to obtain desired speed reductions.

The Thrustor consists of a cylinder in which a piston travels. The cylinder below the piston is filled with oil and a motor driven impeller provides oil pressure in the cylinder to raise the plunger, thus activating the brake release. An adjustment sleeve valve is inserted in the liquid circuit for providing a time delay of the up or down stroke, whenever desired. The impeller motor is a conventional four-pole three-phase squirrel-cage induction motor. If the motor is connected to the line, the piston rises and releases the brake; if the motor is disconnected or the power supply should fail, the motor stops, the up-force at the plunger ceases, the piston returns by its own weight supported by the brake spring in its original position, and the brake is applied.

The Thrustor develops its rated thrust at full speed of the impeller motor. This is the case when the impeller motor is supplied with three-phase alternating current of rated frequency and voltage. If this motor is

Table I—Comparison of Common Systems of a.c. Lowering

No.	A.C. Lowering System	Creeping Speed 64% Negative Load Approx. Full Speed, percent	Current Consumption	Approx. Initial Cost, percent	Safety in Operation	Inching	Comments
1	Oversynchronous lowering	105	Low	100	Good	Fair	No speed reduction below 105 percent except by inching.
2	Multispeed motor	25	Low	150	Good	Good	Squirrel cage motor characteristics. Abrupt acceleration, no gradual speed increase.
3	Mechanical load	25	About 50 percent of rating	130	Good	Fair	Assumed best adjustment of the load brake. Continuous wear requires permanent care and adjustment.
4	A.C. Commutator motor	25	Low	150	Good	Good	Unconventional, custom-built motor type; requires maintenance of commutator and brushes, but extremely simple control by brush shifting.
5	Dynamic braking with d.c.	25	Low	130 to 150	Fair	Fair	For overhauling loads only. The necessary rectifier is an additional device and as such it decreases safety in operation.
6	Counter torque	10	Low	125	Good	Handicap	Light loads might be lifted. Recommended for predetermined loads.
7	Adjustable unbalance voltage	40	High, up to 230 percent of rating	130	Good	Good	The speed reduction at full load is limited. A convenient reduction only obtainable for loads smaller than 50 percent of full load.
8	Variable unbalance voltage	20 with all loads and 30 for light hoist	Line current at lowering can be limited to 125 percent rated	180	Fair	Better than No. 7	The characteristics satisfactory but the addition of apparatus, capacitors, transformers, rectifiers, is not very desirable.
9	Ward Leonard	10 with all loads, best speed adjustment and smooth acceleration	Overall efficiency is low	250	Best	Best	Weight and space requirements of the m.g. set. Care for commutator and brushes. Stand-by consumption of m.g. set is disadvantage.
10	Thrustor	Lowering full load, 25 or 10 Lowering light load, 10 or 5 Hoisting full load, 10 or 5 Hoisting light load, 25 or 10.	Low	125	Best	Best	Creeping speed limited to 10 percent. Creeping speed changes with load. Requires regular inspection.

supplied with a lower frequency and proportional lower voltage, its speed decreases practically with the frequency. Consequently, the liquid pressure developed by the impeller decreases approximately as the square of the speed and in turn, the thrust operating on the brake lever is very rapidly reduced with the speed.

In operation the control device, drum-controller or magnetic control, connects the wound-rotor hoist motor to the line in the lowering direction and on the first lowering point a higher ohmic resistor is inserted in the rotor circuit to obtain reduced motor torque (about ¼ of rated torque). On the first lowering and hoisting position the Thrustor is not connected in parallel with the stator of the motor as usual but across the rotor of the hoist motor. Rotor voltage and frequency are speed responsive. At synchronous speed both are zero and at standstill both have the full rated value. Between synchronous speed and zero speed, rotor voltage and frequency vary inversely proportional to the speed.

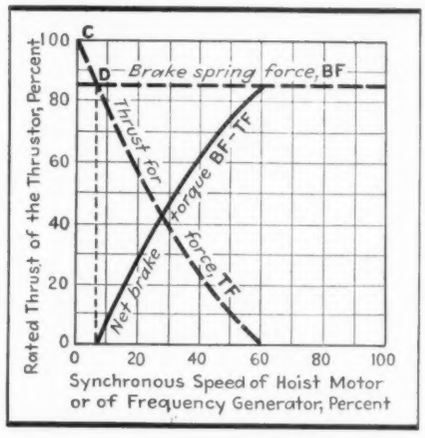


Fig. 2—Thrustor lowering, showing brake force vs hoist motor speed. Brake is entirely released in range CD.

The thrust on the Thrustor *TF*, the force on the brake shoes from the brake spring *BF*, and finally, the force available for brake release *BF-TF* in relation to the speed of the hoist motor is shown in Fig. 2. In the range *CD* where the thrust of the Thrustor exceeds the force of the brake spring, the brake is entirely released. This is the condition shown in Fig. 2 from standstill up to about 7 percent of full

speed. With rising speed the force on the brake shoes *BF-TF* increases gradually with the speed. At about 60 percent lowering speed the full force is applied on the brake shoes. At actual lowering, on the first lowering point the motor speed increases, driven by the lowering load, until the brake torque equals the torque developed by the load plus the torque developed by the hoist motor and an equilibrium is established. At the first hoisting point, this balance is reached when the brake equals the load torque minus the motor torque.

Relatively small changes of the motor speed cause a very powerful increase of the brake force *BF-TF*, as shown in Fig. 2. Consequently, the self-adjusting creeping speed is fairly independent of the magnitude of the load. The self-adjusted creeping speeds on the first hoisting and the first lowering point change with the load from zero to full load are shown in Fig. 3. The creeping speed at hoisting changes from 25 percent at empty hook to 10 percent at full load, and the creeping lowering speed from 10 percent at empty hook to 25 percent at full load. In general this relatively small change in creeping speed is not objectionable even for cranes where a low creeping speed is imperative, as foundry or machine-shop cranes. All loads from empty hook to full load start with the creeping speed on either the first lowering or hoisting point, thus accurate inching is obtained.

The use of the Thrustor regulating brake is not limited merely to lowering; it can be used without any extra cost or additional apparatus for hoisting as well. With the conventional method of using rotor resistors it is not possible to lift all loads with creeping speed on the first controller point, because either the acceleration on light loads is too great or heavier loads do not start at all. Thrustor control allows creeping speed for all loads. As shown in Fig. 2, on the first hoisting point the motor is connected in the up direction, the resistor is selected so that the full load starts positively at the first point. The Thrustor at the first point is rotor connected, thus securing a creeping speed of all loads.

With this lowering system a creeping speed of approximately 20 percent of the full speed is obtainable. If this range is not satisfactory and a wider speed reduction is required, this can be achieved by arranging a separate small wound-rotor induction motor as a frequency converter and using the rotor voltage of this auxiliary motor instead

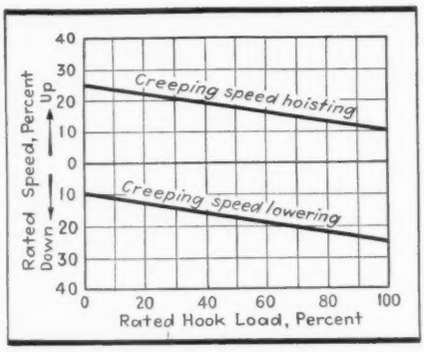


Fig. 3—Thrustor lowering, showing self-adjusting creeping speeds of system.

of that of the main motor for the Thrustor operation. If this auxiliary motor has the same number of poles as the hoist motor and the belt transmission has a ratio of 2 to 1, the frequency obtainable at the slip-rings of the small motor is double that of the rotor of the main motor and the obtainable creeping speed with Thrustor lowering and full load is about 10 percent of full speed.

Tests and experience prove that the burden on the brakes under Thrustor lowering is unexpectedly light. For lowering over greater distances or the entire height, the operator does not use the creeping speed. Rather, he will operate longer distances with over-synchronous lowering during which time the brake is entirely released and will use the creeping speed with the lowering brake where it is needed and this, in general, is only a very short portion of the total life. The difference between the burden on a normal stop-brake and a Thrustor lowering brake actually amounts to only a slightly extended braking period. Experience shows also that there is no appreciable difference in temperature rise, wear and maintenance between a normal hoist stop-brake and a Thrustor lowering brake. In this regard it might be instructive to compare the situation of a Thrustor lowering brake with a mechanical load brake. With the mechanical load brake the braking torque is permanently applied during the entire lowering period, regardless of whether the load is lowered with full speed or with creeping speed, or whether the load-brake torque is needed for speed reduction or not. This results in a permanent wear and power loss.

On the other hand, with Thrustor lowering the lowering-brake is applied only temporarily when needed for creeping speed. This period is relatively short compared with the full lift.

In general the Thrustor lowering system has these characteristics. Without frequency conversion, lowest speed at lowering are: Full load, 25 percent; light loads, 10 percent. With frequency conversion: Full load, 10 percent; light loads, 5 percent. Lowest speed at hoisting: Full load, 10 or 5 percent; light loads, 25 or 10 percent. Low current consumption, relatively low initial cost, extremely safe, and extremely good inching characteristics. The system can be applied readily to existing crane hoists. The Thrustor system is at a disadvantage where: (1) Creeping speed in the downward direction lower than 10 or 25 percent is essential, such as for some high speed hoists; (2) variations of creeping speed at different loads are objectionable; (3) more speed steps between creeping and full speed are important; and (4) inspection and maintenance of the brakes is not done at reasonable intervals during the course of service.

Low Temperature Impact Strength of Cellulosics

From "Some Factors Affecting the Low Temperature Strength of Cellulosic Plastics", by W. E. Gloor, Hercules Powder Company, presented at the annual meeting of the Division of Paint, Varnish, and Plastics Chemistry of the American Chemical Society.

THE IMPACT STRENGTH of cellulose plastics is a good criterion of the mechanical properties of these materials. Tests were made to find how this property would be affected at subzero temperatures by variations in plastic formulation. Ethyl cellulose and cellulose acetate plastics were made and tested at —20 or —40 deg. F. for impact strength. Variations were made in the plasticization, the degree of substitution of the cellulosic, and its intrinsic viscosity (a measure of chain length).

Results indicated that the plasticizer used has a great effect on subzero impact strength, substitution plays a relatively minor part, and the chain length of the cellulose derivative can be important also. The plasticizers giving best impact strength at low temperatures appear to be those the viscosities of which change least with temperature. The use of cellulose acetate or ethyl cellulose of high intrinsic viscosity also aids in raising the impact strength at low temperatures. Testing of samples notched by two different methods indicated that some of the materials become notch sensitive at low temperatures. These results should be of value in formulating plastics for use at low temperatures for design needs.

Basic Standard Drafting Practice Guarantees Correct Interpretation

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Also Chairman of the General Motors Drafting Manual Subcommittee

Advantages and benefits that would be obtained by nation-wide agreement on basic principles of drafting and uniform interpretation of signs, symbols, notes and dimensions. Recommendations for compiling standard master manuals for engineering and drafting departments to promote correct fabrication and inspection of parts.

LACK of definite and clear understanding caused by variances in drafting practices results in many delays, mistakes and set-backs, also much waste and high cost. Because drawings are often misinterpreted, many parts are made that do not meet intended specifications.

Uniformity in basic drafting practices will do much to promote correct interpretation of production drawings. A nationally accepted basic standard of drafting practice would avoid the occurrence of much waste of material and time.

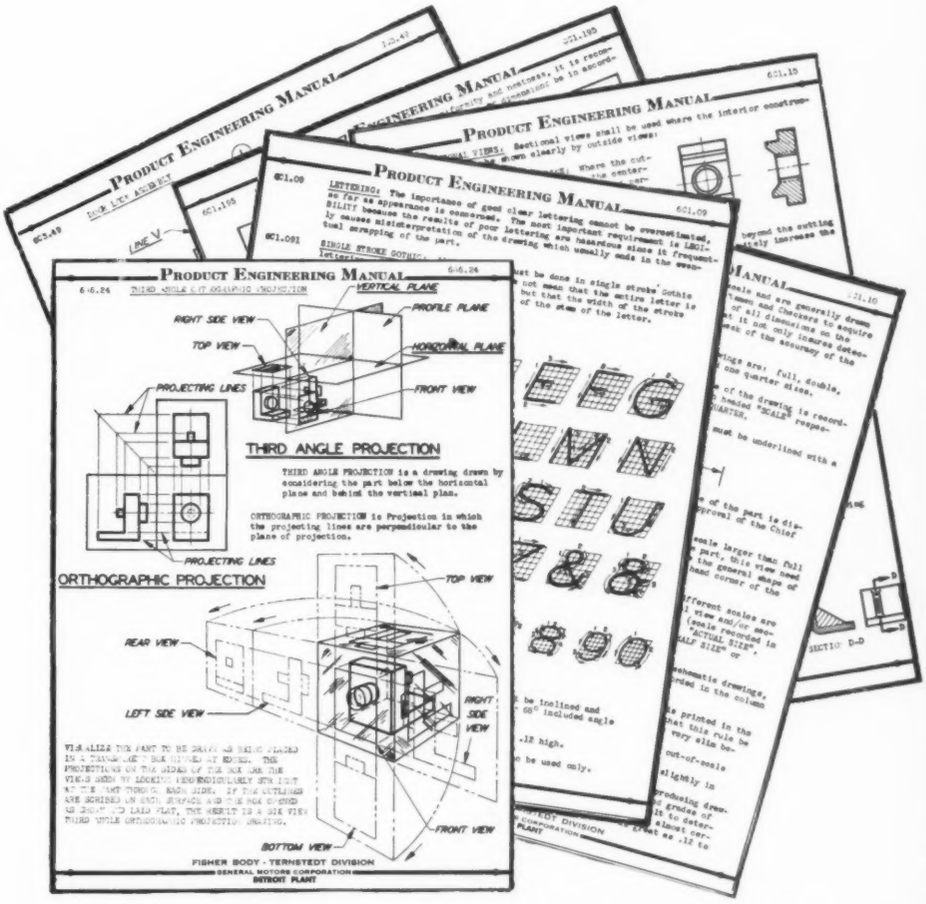
Drafting standards have been established by several organizations such as The American Standards Association, The Military Services, and The Society of Automotive Engineers. But a long-range program must be formulated and adopted before a national basic uniformity is achieved.

The fear that such standardization means static crystallization is not well founded. Industry is dynamic. With basic principles clearly defined, an over-all plan of standardization can be set up that is both flexible and adaptable enough to meet new problems as they arise.

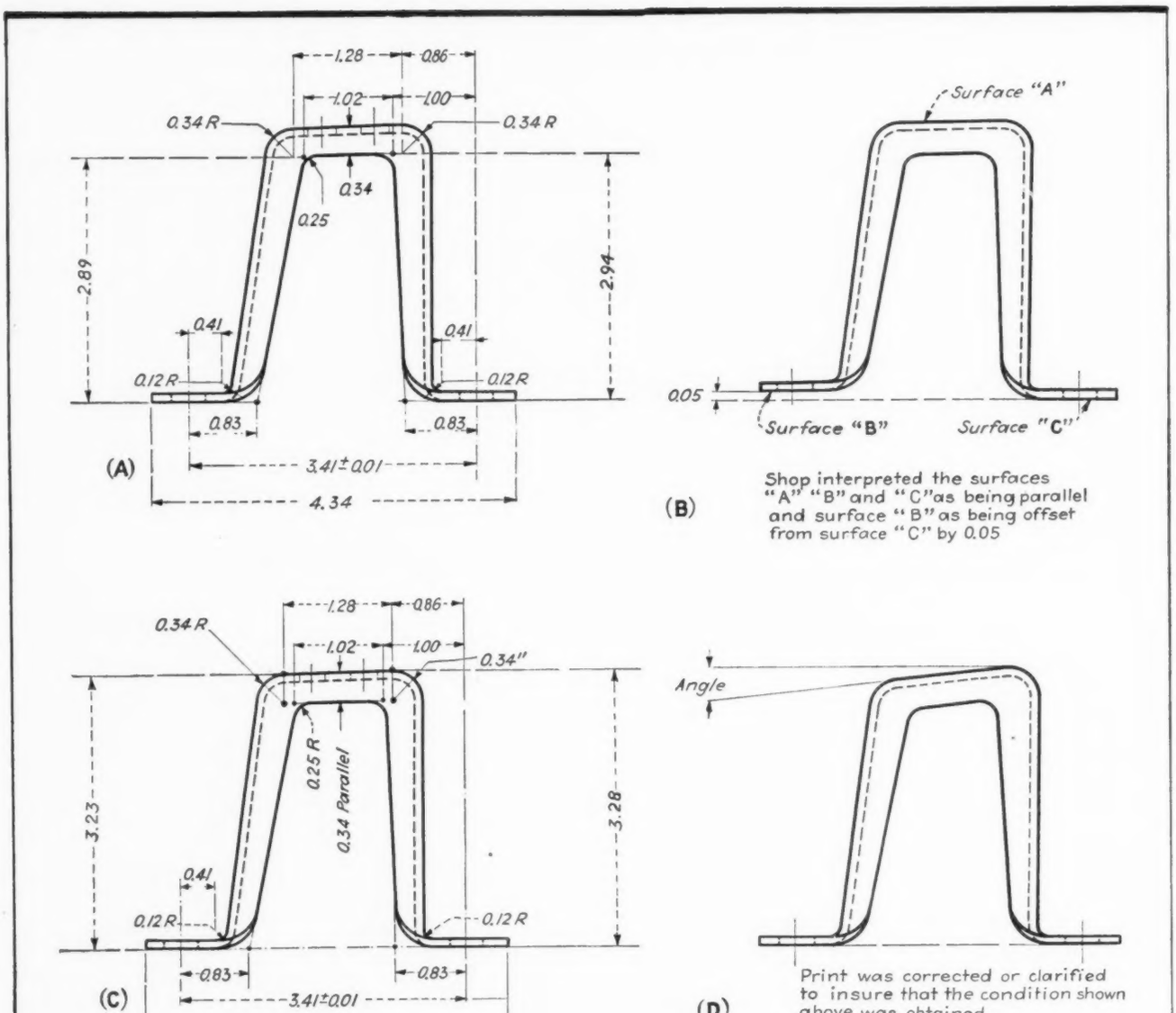
The ideal objective is to create a text book of basic standards comparable to Webster's International Dictionary, which has become a standard in America on elements of our language. To be effective such a book would require endorsement by the national engineering societies and the several large industrial groups.

Drafting practices especially adapted to the individual problems of a specific organization need not be hampered, since these inter-organization practices can be included in the scope of the nationally approved basic standards. Experience proves that the use of a manual by an organization means better production of a better product with resultant economy.

The first step is to reconcile conflicting ideas within each type of industry in the same manner that the aircraft engine and accessory manufacturers have established an accepted drafting standard for their work. After each type of industry has established its drafting standard, it will be opportune for the different groups to get together and agree upon



Sample pages from an engineering manual. Combination alphabetical and numerical system of numbering paragraphs eliminates page numbers.



Deficiency of dimensions on production drawings is cause of much waste. (A) Original drawing of a formed part. (B) Shops interpretation of original drawing. (C) Corrected drawing. (D) Part as desired by engineering department.

basic standards. Each type of industry will join the national body with the full understanding that its tentative inter-organization standards will be subject to modifications to bring them in line with approved basic practices. Any material common only to the work of one type of industry can be treated as special and handled by a subcommittee.

Out of the combined cooperative efforts of all these groups eventually will come the first transcript of a master manual. This master manual will be divided into sections, with the main section devoted entirely to basic principles of drafting such as projection, arrangement of view, lines and line work, dimensioning, conventional symbols, and lettering. The divisions would be similar to ASAZ14.1-1946.

The ultimate revision of this transcript will be accepted as a national standard by all types of industry. The subsections, covering special practices common to only one type of industry, will be approved by each of these special groups. Such groups will include farm equipment, instruments, ships, fabricating machines, aircraft engines and accessories, airplane, automotive, automobile bodies, automobile hardware, electrical equipment, construction equipment, and others.

These special subsections will be so devised as to come within the broad elementary provisions embodied in the main section of basic principles. As a result, the various manufacturers will have a drafting guide that will not only meet their own requirements but will be in complete accord with basic principles in current use throughout the nation.

Benefits of Basic Standards

This nation-wide agreement on basic principles will mean a common drafting room language that is understandable to all persons concerned with manufacture, sub-contracting, selling, or replacement since there will be uniform interpretation of signs, symbols, presentations, and dimensions. Educational institutions

will also benefit, as they will be in a better position to coordinate their mechanical drawing curriculum.

The standard master manual here proposed will have to be designated as the official and final authority in so far as the correct reading of mechanical drawings is concerned. It must be authoritative to the degree that it can be used as a reference in litigations to settle disputes pertaining to the correct interpretation of mechanical drawings. A drawing is a contract, and should be just as binding from an engineering standpoint as any legal document.

Experienced engineering executives agree that standard drafting practice has both merit and value, but some feel that the benefits are evanescent and remote. From a practical point of view, only one criterion can be used to appraise the value of a standard drafting practice, that is, its economic effects.

All the savings and increased profits that accrue from strict observance of a definite standard cannot be included in a short summary, but some of the more important are:

1. Less Repetition of Work. When parts are designed by one concern and manufactred by another, two or more sets of drawings are not necessary. Duplication of effort and repeated research are avoided because prior solutions to recurring problems are available immediately.

2. Reduces Supervision. Draftsmen do not need step-by-step guidance.

3. Fewer Arguments. Discourages time-wasting disagreements caused by conflicting ideas.

4. Fewer Costly Errors. Establishes a common drafting room language, thus promotes correct interpretation of presentations, dimensions, conventions and symbols.

5. Facilitates Manufacture. Design data shows draftsmen how parts should be designed for economical production.

6. Standardizes Tools and Materials. Insures whenever possible the use of existing tools, gages, standard stock sizes of materials.

7. Insures Interchangeability. Pertinent data are tabulated covering fits, tolerances and surface finishes.

8. Simplifies Routine. Standardized part names simplify compiling and alphabetical filing of parts lists.

In every organization problems exist that are peculiar to the work in hand. The writer of a satisfactory inter-organization manual must, of course, be well acquainted with the source and solution of such problems.

When highly skilled personnel use the manual, the fact should influence the context of the book. When the personnel is comparatively inexperienced, the presentation should be elementary and detailed. When practicable, even data published in standard reference books should be included.

A properly prepared engineering manual should enable any engineer or draftsman to find desired information quickly and easily. The manual should tell how any drafting operation is correctly performed in accordance with approved uniform practice.

In the absence of uniform basic standards, disagreement frequently arises as to how a specific drawing should be made. For example, one engineer might indicate that the location of holes, keyways or slots, unless otherwise specified are to be within 0.005 in. of a common centerline; another engineer might indicate 0.001 in. is the proper tolerance, or that they can vary freely. Such conflicting and sometimes inaccurate statements reflect unfavorably on the efficient administration of the engineering department.

Much uncertainty and lack of agreement on the interpretation of production drawings can be avoided by standard practices. But often it is almost impossible to indicate on one component production drawing all the dimensions best suited for each particular machining and process operation. And yet, a production drawing should be regarded as completely satisfactory only when it includes all dimensions, notes, and data, necessary for manufacturing departments to fabricate the part completely and correctly and to inspect the part.

Dimensions used by manufacturing departments may or may not be the same as those specified on the drawing, but are satisfactory so long as the finished part measures up to the specifications and requirements presented on the drawing. Again, manufacturing departments might decide to use dimensions better suited for performing a special machining operation and constructing tools and gages than the dimensions shown on the drawing. Of course, to do this they would have to understand the requirements of the part from a functional standpoint, and here the uniform standards would play a basic part.

Engineers and draftsmen often disagree on the exact definition of some technical terms. Text books give different definitions and this results in ambiguity. Technical terms originate from necessity when a new symbol is needed for precise meaning. A technical idea may be too complicated to express in common terms, or require too many words if a single symbolic word is not found to express it. This conflict could be standardized by the adoption of a master manual. A section on definitions of technical terms should be included, flexible enough to add new terms as they come into use.

Engineering drawing is the language of a profession, just as mathematics is the language of science. Since engineering drawing consists of the portrayal of ideas by means of lines, words and terms, these elements must convey definite concepts.

Although many concerns in the automotive and aircraft fields have excellent drafting manuals, a close examination will show a lack of system in the writing of drawing notes. It is

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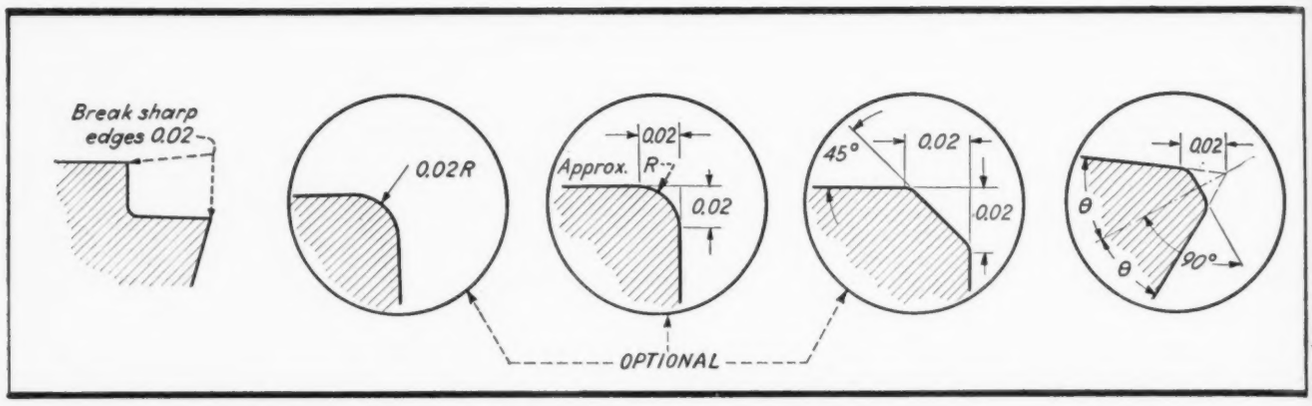
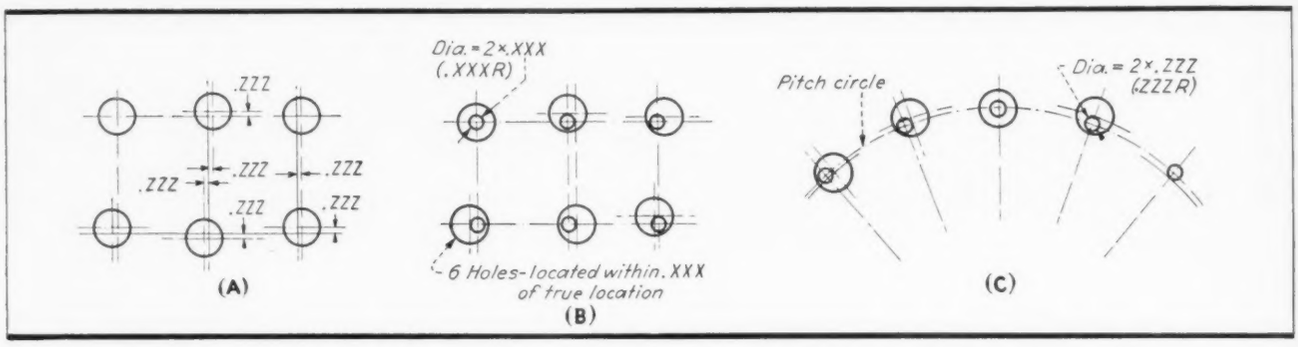
Abstract from American Standard Drawing and Drafting Room Practice (ASA Z14.1-1935)

"It is not always safe to say, that the most used is the best practice, but whenever a definite majority prefers one method and the balance is made up of many small minorities, it may generally be assumed that the best practice is that of the majority. When two or more methods have nearly the same number of followers, the one coming into increasing use by progressive firms has been given preference over one declining in use."

"Each of the variations referred to has its advocates and it is recognized that a good many will be reluctant to change from some method they may be using to another which (to them) seems to be no better, but it is hoped that in the interest of uniformity the adopted standard method of representation may be acceptable by them in spite of individual preference. Only by such concessions can a standard practice become a reality."



A sharp edge may be broken in the form of a radius, an approximate radius, or a chamfer, as long as the method employed produces an edge that is free of burrs and is not sharp.



All holes shown in (A), (B), and (C), are within the tolerance plus or minus ZZZ of true position.

not practicable to list all drawing notes used by various industries in one volume because of the wide differences in the nature of their respective work. It should be recommended, therefore, that all persons receiving the master manual furnish a list of all their acceptable notes used.

The drawing notes accepted as standard should be listed in three groups; (1) General Drawing Notes —notes that apply to the part or assembly in general; (2) Local Drawing Notes—notes that are commonly used and that apply to a local or particular portion of the part or assembly; (3) Special Drawing Notes—notes not generally used and that are usually limited to one special application. It is advisable to show illustrations of the application and usage of special drawing notes, and thus simplify the task of finding the note required and insure correct selection.

Rules should also be given for writing notes. Notes should not duplicate information recorded in title block or repeat given dimensions. Operational terms, nonessential words and quotations marks should be avoided.

The Dewey decimal system commonly employed by libraries for indexing, is highly recommended in numbering the manual pages. The basic idea of having a combination alphabetical and numbering system, consisting of both whole and decimal numbers with each paragraph numbered individually using combination numbers, that is, 1A1.01, 1A1.02, 1A1.03, and so on, does away with page numbers thus permitting the addition or withdrawal of material without the necessity of changing or rearranging page numbers. The first paragraph of each page is repeated in the lower right or left hand corner of the sheet and considered as the page number. Therefore the first paragraph number and the page number are synonymous, which makes it possible to add or remove any number of pages without introducing any confusion or disrupting the numbering sequence.

ACKNOWLEDGEMENTS—The author acknowledges the helpful suggestions and encouraging comments given by C. E. Hilton, staff engineer, American Standards Association; M. L. Stoner, staff engineer, Society of Automotive Engineers; G. F. Nordenholt, editor, *Product Engineering;* and A. J. DeSana, chief engineer, Fisher Body-Ternstedt Div., General Motors Corporation.

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DURING THE LAST CENTURY brewing has become recognized as one of the world's major industries. It ranked seventh in the United States in 1940. Scientific progress and education, electricity and modern equipment have contributed to this increased production.

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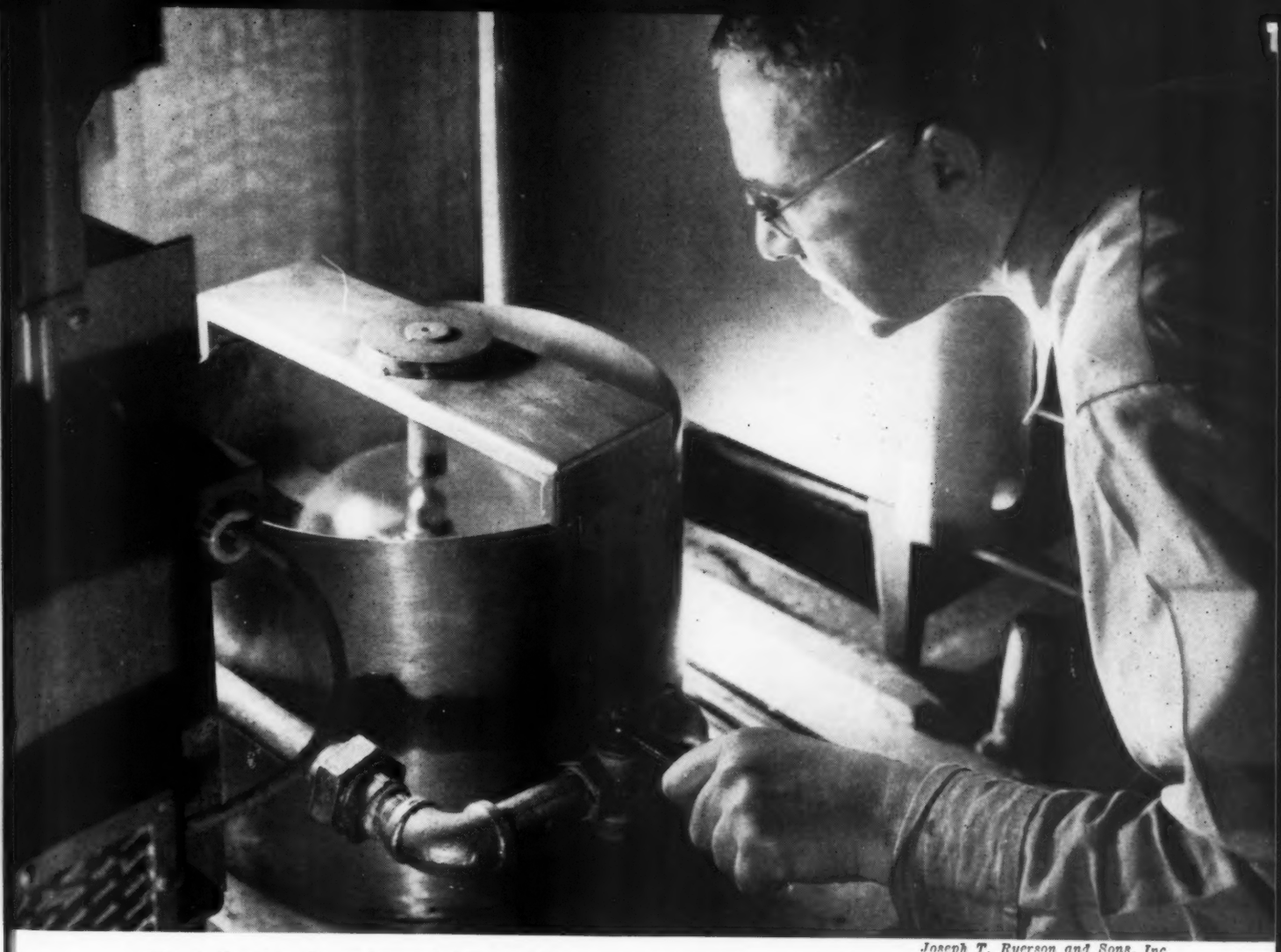
ONE OF THE FIRST PUMPS to be completely mechanized was the tympanum of the ancient Egyptians. It was used to pump water from streams to the irrigation ditches of the nearby fields. The tympanum was a large wheel with paddles built around the periphery. Four spiral troughs were mounted on one of the faces of the wheel. The lower part of the wheel was placed in the stream where the flow of water turned it around. Water was picked up and trapped by the troughs as the wheel turned. Further turning raised the water to the hollow, central hub, where it ran off to the ditches.



Joseph T. Ryerson and Sons, Inc.

Fig. 1—Running the end-quench hardenability test. Test specimen is held vertically in the fixture.

Hardness and Hardenability In Carbon and Alloy Steels

Difference between hardness and hardenability is explained. The method of running standard end-quench hardenability tests is outlined. The effect of carbon and other alloys on hardness and hardenability are discussed. Typical hardenability curves are shown.

ONE significant advance in metallurgy, which is of benefit to the design engineer, is the evaluation of the effects of alloying elements in steel in terms of hardenability.

Because of the similarity of the terms "hardness" and "hardenability", their meanings are often confused. "Hardness" is a measure of relative resistance to deformation by indentation. The hardness of steels is determined on testing machines such as the Brinell and Rockwell hardness testers. By reference to empirical tables, hardness values can be translated into tensile strength values.

The tensile strength in pounds per square inch of ductile steels is roughly 500 times Brinell hardness. But the hardness test tells nothing about the ductility, toughness, yield strength, fatigue strength, wear resistance, and machinability of a steel. Two steels of the same hardness may have equivalent tensile strengths, but their other mechanical properties could differ greatly.

The maximum hardness that can be obtained in any steel depends upon its carbon content and not upon alloy additions. The relation between carbon content and maximum obtainable hardness is shown by Curve (*A*) in

Fig. 2. These hardnesses are obtained when the steel is quenched to a fully martensitic structure such as that shown in Fig. 3. Curve (*B*) and shaded zone (*C*), Fig. 2, show the hardnesses of 50 percent martensitic structures in carbon and alloy steel respectively.

The "hardenability" of a steel of any given carbon content is a measure of the diameter in inches of the maximum size round bar that can be quenched to martensite throughout and thus produce maximum average hardness for a steel of that carbon content. The maximum diameter of steel bar that can be quenched throughout to martensite, even under the fastest obtainable quenching speeds, depends upon its carbon and alloy content. The alloys slow the rate of decomposition of austenite. In an alloy steel and a plain carbon steel having the same carbon content and of the same size, when quenched under similar conditions the depth to which martensite can form will be much greater in the alloy steel than in a plain carbon steel. Fig. 4 shows the differences in hardness gradient between an alloy and a carbon steel of the same carbon content.

Hardenability can be appraised by one of two methods: (1) the hardness from a given distance from the water cooled end of the specimen, or (2) by the depth of martensitic hardening. The usual end-quench test is generally not suitable for shallow hardening steels such as plain carbon steels, nor is this test generally considered satisfactory for application to heavy sections.

In the Jominy test a specimen with dimensions as shown in Fig. 5(*A*) is normalized, held for 20 to 30 min. at the proper quenching temperature for the steel under test and quenched by a stream of water impinging against its one-inch diameter face. During quenching, the test bar is held vertically in a fixture such as is shown in Fig. 5(*B*) that holds the bar central and at a distance of ½ in. above the half-inch diameter orifice. The water is kept at 75 ± 5 deg. F. and the flow is regulated by a quick-acting valve that permits a 2½ in. free water rise when the stream is not impeded by the test bar.

After quenching in the fixture for 10 min., the test bar is cooled by complete immersion. After cooling, two flats are ground, 180 deg. apart and 0.015 in. deep, along the length of the bar, and then Rockwell hardness readings are made at 1/16 in. intervals along the center lines of the flats.

Because only one face of the test specimen is subjected to the quenching action of the water, the speed of cooling ranges from practically instantaneous at the quenched end of the specimen to a slow rate at the opposite end. The hardness value at any point along a test specimen depends upon the cooling rate at that point and the hardenability of the steel being tested.

Typical curves of the relationship of hardness to distance from the quenched end of the test bar for several types of 0.40 percent carbon and one 0.30 percent carbon steel are shown in Fig. 6. Such curves are usually plotted on a standard form that gives the cooling rates in deg. F. per sec. at 1,300 deg. F. at various points along the test bar. Curve (*A*) is that for a highly alloyed steel in which the rate of decomposition of austenite is so slow that martensite can form even at slow cooling rates. Curves (*B*) and (*C*) are typical of steels of medium alloy content; these steels maintain peak hardness for a certain distance along the bar and then decrease suddenly in hardness. Curves (*D*) and (*E*) are typical of plain carbon steels; these steels have a peak hardness at the quenched end that quickly drops off to low values.

All of the 0.40 C steels mentioned, as shown by the curves, reach the same maximum hardness of Rockwell C-60 at the quenched end. The differences in alloy content cause differ-

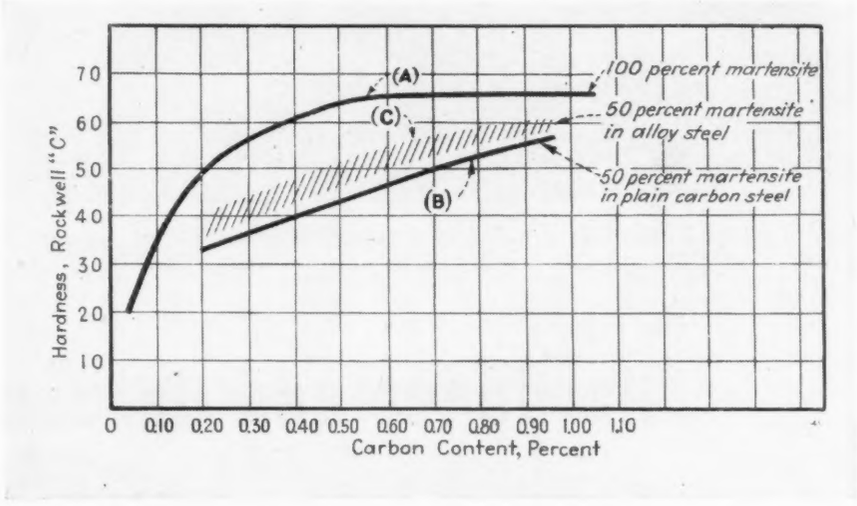


Fig. 2—Relation between carbon content and maximum obtainable hardness in carbon and alloy steels, when quenched to a fully martensitic structure.

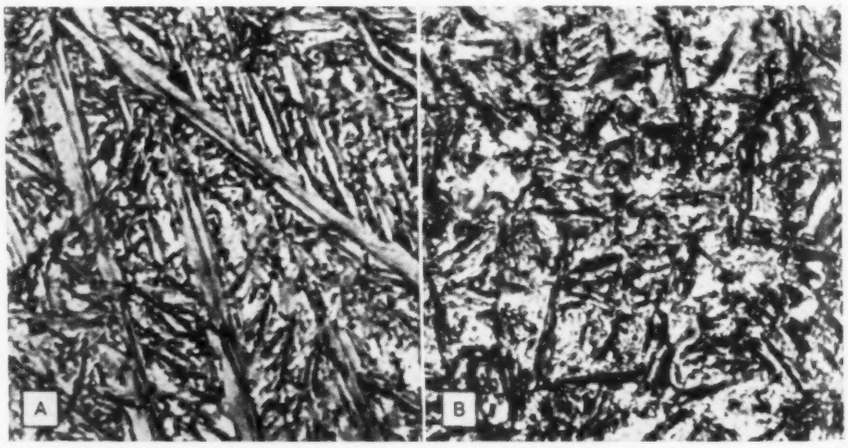


Fig. 3—Martensitic structure required to produce full hardness.—(A) 100 percent acicular martensite in a 1095 grade steel. 2,000 deg. F. end quench in water. (B) 100 percent martensite in a 4640 grade steel. 1,550 deg. F. water quench.

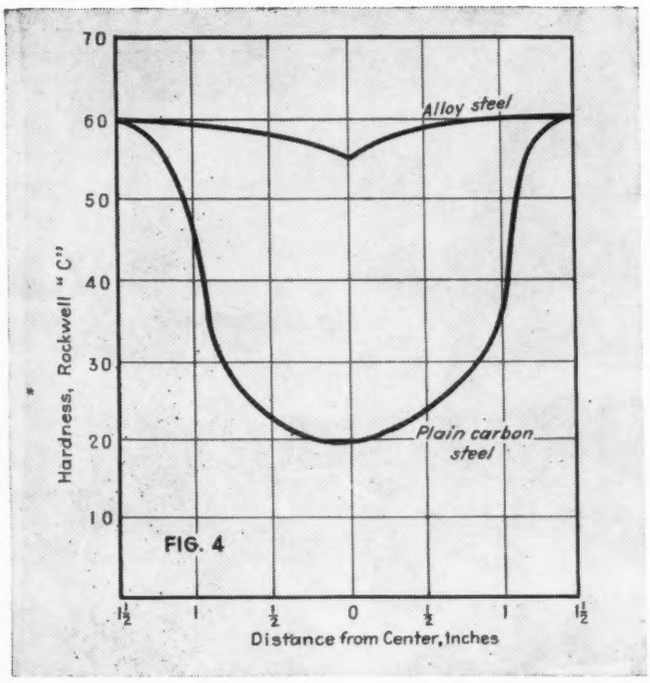


Fig. 4—Hardness gradients across 3 in. dia. bars of a carbon and an alloy steel, each containing 0.40 percent carbon.

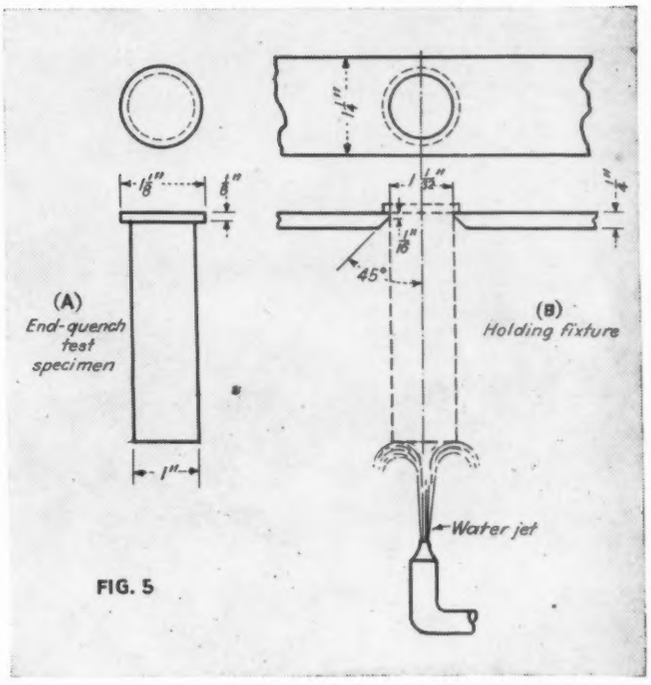


Fig. 5—Dimensions of end-quench test specimen and holding fixture. End of test bar is ½ in. above water jet orifice.

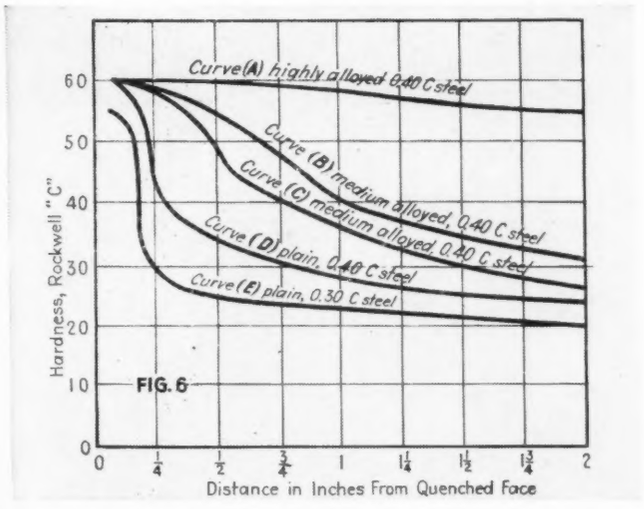

Curve (A) highly alloyed 0.40 C steel
Curve (B) medium alloyed, 0.40 C steel
Curve (C) medium alloyed, 0.40 C steel
Curve (D) plain, 0.40 C steel
Curve (E) plain, 0.30 C steel
Hardness, Rockwell "C"
FIG. 6
Distance in Inches From Quenched Face

Fig. 6—Curves showing relation of hardness to distance from quenched end of bars of five types of carbon steel.

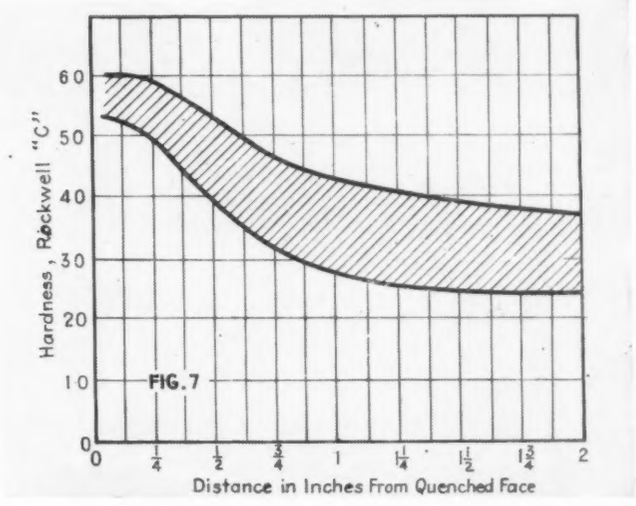


Fig. 7—Hardenability band of A.I.S.I. A-8640 steel. This band is typical of an alloy steel of medium harden-ability.

ent rates of drop beyond the quenched end. Curve (*E*), a 0.30 percent plain carbon steel reaches a maximum hardness of Rockwell C-55 and shows the same general rate of drop as the 0.40 percent plain carbon steel.

In selecting the proper steel for a given part, hardenability data permit choosing the type of steel with the least alloy content that will give the required tensile strength. Comparisons of hardenability are used to determine permissible substitutions of one type of steel for another in established designs.

The so-called H steels, which are part of SAE or AISI analyses, guarantee a definite hardenability band for the different types of steel. The hardenability band of AISI A-8640 steel is shown in Fig. 7. For example, when an SAE grade of steel having an H prefix is ordered, this means that slightly wider ranges are permitted in chemical analysis, because such small variations in composition are not important provided the specified hardenability is present. The chemical analysis should be specified even though hardenability is also specified, since certain physical properties, and fabrication and heat-treating procedures are dependent upon the composition of the steel.

Since the best combination of impact and tensile properties are obtained when the microstructure is composed of tempered martensite, steels used in highly stressed parts should have sufficient hardenability to permit the formation of martensite that can be subsequently tempered to the desired tensile strength. Differences in steel-making practices and the presence of incidental alloys in different analyses can cause marked differences in hardenability in steels of the same chemical analysis range.

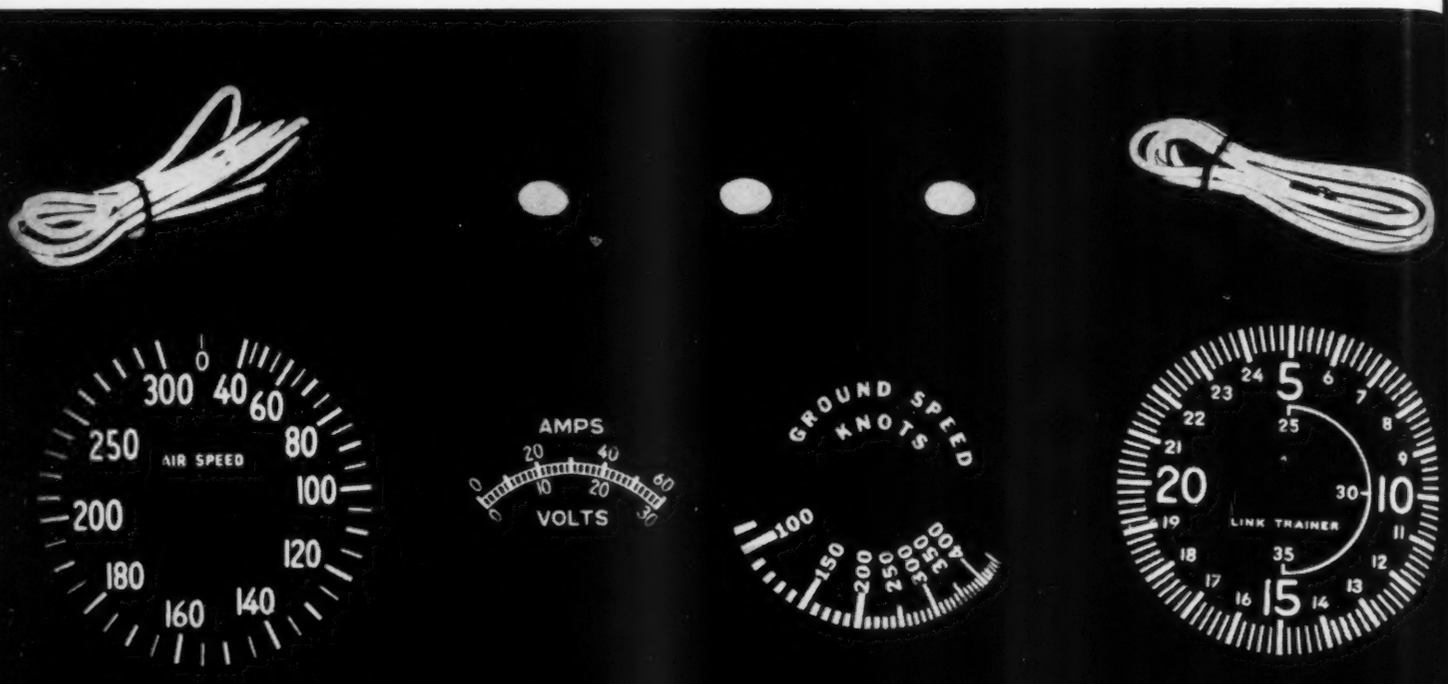


Fig. 1—Radioactive rope, luminous deck and personnel markers one inch in diameter, and airplane instrument dials treated wih fluorescent paint, can be seen clearly in darkness.

LUMINOUS MATERIALS

Properties and Typical Applications

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Materials and methods used to produce luminescence without a power source. Brightness, eye sensitivity and applications of phosphorescent, fluorescent and self luminous materials, as well as basic principles of photometry, are discussed.

NIGHT operations were aided during the war by phosphorescent, fluorescent and self luminous radium-activated materials. Experience with these materials indicates applications in machine design and other phases of industry.

Substances that become luminous when exposed to light, and continue to glow for a period of time after the source of activation has been removed, are called phosphorescent. Other substances that glow brightly in the visible part of the spectrum while activated by ultraviolet light are known as fluorescent. Certain inorganic sulphides glow continuously when a small amount of a radioactive salt is mixed with them. These are called self luminous. Table I shows some of the common types of luminescence, as well as methods for producing them.

Phosphorescence and Fluorescence

The dividing line between phosphorescense and fluorescence is not well defined since all luminous materials can have a finite luminous life, although it may be of extremely short duration. In fluorescence there is a spontaneous transition of the molecule from an "excited" state to a lower energy level; in phosphorescence a portion of the excited molecules pass to a metastable state of somewhat less energy, without emitting light. They can be made to emit light when energy is restored to them, bringing them back to the excited state. This energy comes from the heat of the surrounding medium. Time of return to the energy level at which they emit light depends on temperature.

In fluorescence the mean life of the

excited state is in the order of microseconds, while the excited state in phosphorescence can be a matter of hours. Fluorescence and phosphorescence are monomolecular reactions and the decay of intensity of radiation follows the exponential law,

$$I = I_o e^{-at},$$

where I is the intensity at any time t, I_o the original intensity and a the transition probability. Experimental proof of this law has been obtained.

Luminous substances can be divided into three types. Organic compounds, pure inorganic compounds, and inorganic crystals activated by inorganic impurities. The list of fluorescent organic compounds is long; in general they include many organic compounds with large molecules containing the benzene ring. Among the most useful are many organic dyes, such as fluorescein, rhodamines, eosin, thiaflavin and safranine. A large number of aromatic hydrocarbons and many of the common fats and oils are also fluorescent. The fluorescent properties of organic materials can be modified by the substance in which they are dissolved. A highly fluorescent material in one solvent may be weakly fluorescent in another. The knowledge of the fluorescence of organic compounds is in an uncertain state and much of it is contradictory.

Of the inorganic crystalline salts capable of luminescence, the sulphides of zinc, calcium and strontium are among the most important, since they are capable of a long afterglow. Some of the silicate and tungstate phosphors, which are characterized by an afterglow of short duration and a high luminous yield, are used in fluorescent lamps and screens.

Measurement of Brightness

Evaluation of properties, uses and causes of deterioration of luminous materials depends on the measurement of brightness and color. Because of the low luminosity of the radiation from phosphorescent and fluorescent substance, these measurements are difficult and do not have a high order of accuracy.

Certain units of measurement and terms used in photometric work should be defined. The most fundamental notion of photometry is that of luminous intensity. This term designates the strength of the source of radiation; the unit is the "international candle." This unit has been arbitrarily chosen by the International Committee on Photometric Units as a fraction of the average intensity of a set of electric lamps operated under given conditions. Intensity of the source is measured in candle power. Radiation from the source in the visible region of the spectrum is called "luminous flux" and the unit is the "lumen," the flux emitted in a unit solid angle by a one candle power source.

Phosphorescent and fluorescent substances are usually described in terms of brightness and color. Brightness is the quotient of the luminous intensity in the direction under consideration and the area of the luminous surface projected on the plane perpendicular to this direction. The unit of brightness is the "lambert," which is the brightness of a body emitting one lumen of flux per centimeter. This unit is inconveniently large, so that a unit one millionth of this magnitude is often used. A perfectly reflecting and diffusing surface illuminated by 1 candle power source at a distance of 10 meters has a brightness of 1 "microlambert." Another common unit of brightness is the "foot-lambert" which is the brightness of a body radiating one lumen per square foot. To change foot-lamberts to microlamberts, multiply by 1076.4; to change microlamberts to foot-lamberts, divide by 0.00929. The following is the brightness of some commonly observed objects:

Object	Brightness in microlamberts
Concrete highway under approximately full moon	8.5
White paper in full moonlight	10.0
White paper in full sunlight	10^7
Radioactive personnel marker used by the Navy	10
Luminous tape 30 minutes after a 1 minute activation by tungsten lamp illumination of 500 ft. candles.	0.1
Weather deck of a ship painted gray in full moonlight	0.1

Eye Sensitivity

The "distinctness" with which an eye can perceive a luminous object depends on several factors: Size and shape of the object, brightness, contrast with the background, visual characteristics of the observer's eye, and the state of "dark adaptation." The larger the luminous surface the more easily and farther it can be seen. However, if the surface is a number or letter, in order to be legible, not only has the area to be of sufficient size, but neighboring portions of the character must be enough separated to be resolved by the eye. For the normal eye, angular separation of two luminous points must be at least one minute of arc in order that the eye perceive them as separate sources.

When passing suddenly from a

Table I—Methods of Producing Luminescence

Type	Method of Excitation	Substance Capable of Excitation
Thermo luminescence	Heating below redness	Minerals rich in calcium carbonate
Tribo luminescence	Rubbing, breaking or crushing	Many organic crystals some inorganic minerals
Cathode luminescence	Bombardment by stream of electrons or cathode rays	Fluorites, arogonite, willemite and many others used in oscilloscope screens
X-ray luminescence	Excitation by X-rays	Rock salt, fluorite, platimo cyanide, zinc sulphide, etc., used in X-ray diagnosis
Electro luminescence	Application of high potential discharge	
Crystal luminescence	Crystallization of certain salts	Sodium and potassium chloride, bromide and iodide
Chemi luminescence	During chemical reactions	Reaction of oxides of calcium, barium or potassium with sulfuric acid
Bio luminescence	Production of light by living organisms	Light given out by animals during life processes: "phosphorescent" wood.

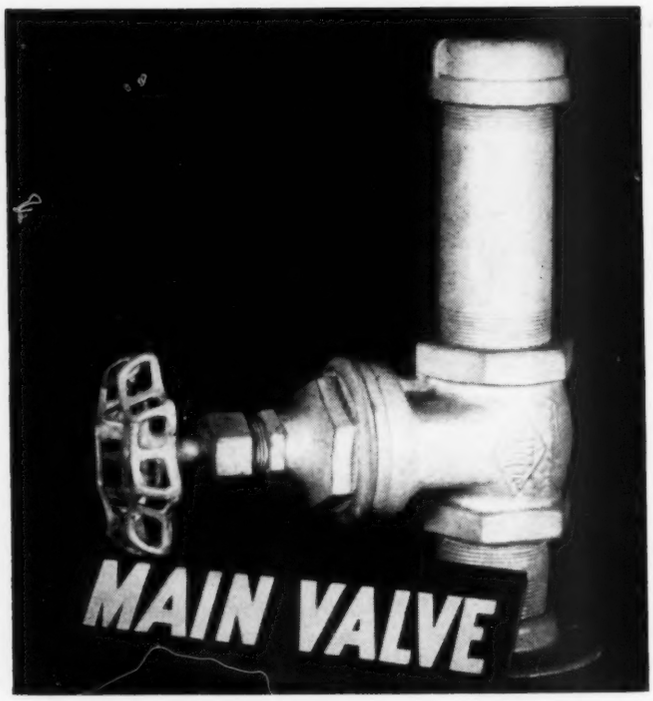

Fig. 2—Valve treated with phosphorescent zinc sulphide paint can be seen in total darkness, and eliminates need for power sources at inconvenient locations. Zinc sulphide is stable to the action of heat, light and moisture.

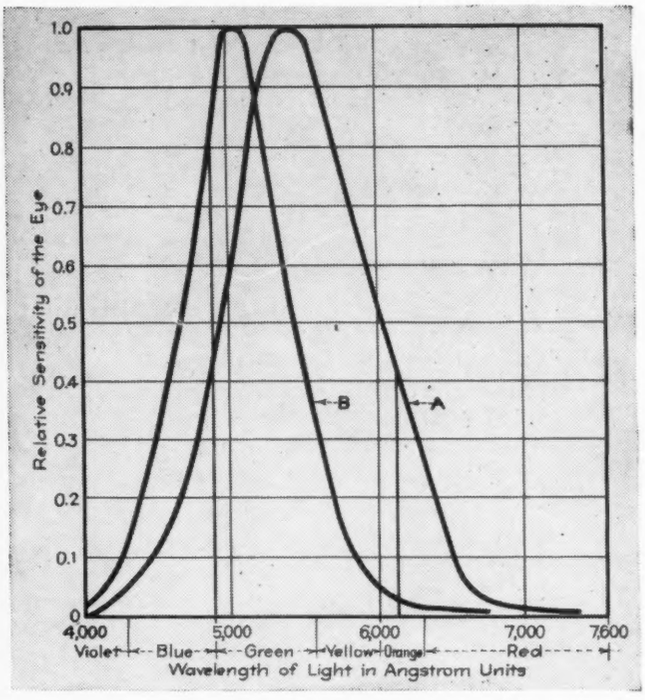

Fig. 3—"Dark adaptation," or increased sensitivity of the eye to light which takes place when entering a dimly lighted room, is characterized by a shift of the sensitivity curve from normal (A) toward the blue end of the spectrum (B).

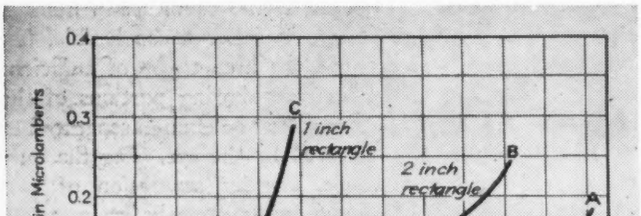

Fig. 4—Distance that phosphorescent calcium sulphide areas can be seen by a fully "dark adapted" eye, against a black background at brightnesses below 0.5 microlamberts.

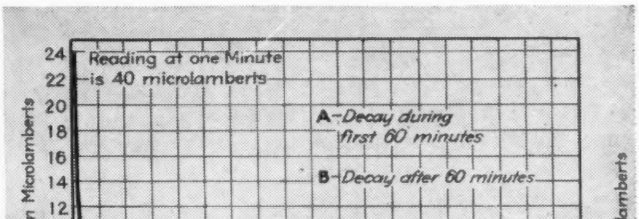

Fig. 5—Decay of brightness of phosphorescent calcium sulphide adhesive tape. In evaluating this material, scale at left applies to curve A and right scale to curve B only.

region of high light intensity into a dimly lighted room, the eye can see only indistinctly. However, the eye gradually becomes more and more sensitive and after some minutes can perceive objects much more plainly. This phenomenon is known as "dark adaptation" and the degree and speed of adaptation varies with the individual. It is characterized by a rise in the sensitivity of the retina of the eye to light, which is slow during the first 5 minutes and more rapid during the following 20 or 30 minutes. After this there is a slow rise to maximum sensitivity. An extremely short exposure to bright light after dark adaptation causes a slight temporary decrease in the state of dark adaptation. During the period that the eye is becoming dark adapted, and there is an increase in sensitivity, there is at the same time a relative increase in sensitivity in the blue region of the spectrum compared to the red; the sensitivity curve of the human eye shifts toward the blue end of the spectrum. This phenomenon is shown in Fig. 3, as the illumination decreases to a low value. Curve *A* shows the maximum sensitivity at about 550 Angstroms for the eye at normal levels of illumination, the "photopic" state. Curve *B* shows the shift of the point of maximum sensitivity to 5070 Angstroms for the fully dark-adapted eye. Thus, in selecting phosphorescent or fluorescent materials for any specific use, the state of dark adaptation of the eye of the observer must be considered.

As the eye is subjected to decreasing intensities of light at the red end of the spectrum, the sensation of light and color are lost simultaneously; but

as the wave length of the light decreases the sensation of color is lost sooner than the sensation of light. The disappearance of color occurs at a brightness of approximately 1.0 microlambert.

Phosphorescent and Fluorescent Legibility

Legibility of luminous characters of low brightness depends on the contrast between the letter or numeral and the background, which in turn depends on the amount of illumination present, on the reflection coefficient of both the background and luminous material and on the brightness of the material itself. Where the reflection coefficient of the material is high, as in uncovered metal surfaces, the brightness of the background in low illumination may be greater than the brightness of the self luminous material. Where luminous material is used, the background is usually a material of a low coefficient of reflection; yet when illuminated with light from a full moon its brightness may be as high as 5 microlamberts. The brightness of the luminous material must be at least 30 percent greater than the background for luminosities in the neighborhood of 2 microlamberts and at least 10 percent greater for luminosities of 8 to 10 microlamberts.

Four examples of the legibility of characters made from commercially available phosphorescent and fluorescent materials are given in the following paragraphs. Characters were read in a dark room against a black background by observers with normal vision, after remaining in total darkness for 45 minutes.

EXAMPLE 1. Legibility of characters made from self luminous radioactive zinc sulphide is shown in Table II. The characters are block type numerals, with dimensions approximately those of the markings of standard compass cards, engine telegraph dials, clock and gauge faces and airplane instrument dials.

EXAMPLE 2. In using phosphorescent tape to aid operations during power failure, in darkened spaces such as engine rooms, for evacuation of personnel from inside of a ship or similar activities, it is necessary to know at what distances the human eye can see luminous areas made from commercial tapes. Fig. 4 shows the distance in feet that 3 luminous rectangles, 6 in. long and 1, 2 and 3 in. wide, can readily be seen by a fully dark adapted eye after the material has reached a relatively low brightness. The luminous material is calcium sulphide, which radiates in the blue region of the visible spectrum. Since phosphorescent material must be activated by some source of light and since the brightness of the material gradually dies away after activation ceases, it is necessary to know this decay in order to evaluate the material.

Fig. 5 shows how to make this calculation. Suppose one wishes to know how long a rectangle of calcium sulphide tape, 3 in. wide and 6 in. long, will be readily visible in total darkness at 20 feet. From curve *A*, Fig. 3, it is seen that the minimum useful brightness is approximately 0.09 microlamberts. From curve *B*, Fig. 4, it is seen that the tape will reach this brightness approximately 260 min. after activation has ceased.

Table II—Legibility of Numerals Made From Radioactive Zinc Sulphide

Dimensions of letters, millimeters		Brightness microlamberts	Distance at which numeral is readable centimeters
Height	Width		
3	0.25	4	not legible
5	0.50	4	not legible
7	1.00	4	not legible
10	1.0	4	34
13	1.5	4	75
16	2.0	4	98
19	2.0	4	115
22	2.5	4	139
25	3.0	4	158
3	0.25	6	not legible
5	0.50	6	not legible
7	1.0	6	not legible
10	1.0	6	34
13	1.5	6	75
16	2.0	6	120
19	2.0	6	138
22	2.5	6	157
25	3.0	6	200
3	0.25	9	not legible
5	0.50	9	not legible
7	1.0	9	37
10	1.0	9	83
13	1.5	9	113
16	2.0	9	145
19	2.0	9	175
22	2.5	9	208
25	3.0	9	234

EXAMPLE 3. Figs. 6 and 7 give the distances that letters and circular spots can be seen by the dark adapted eye against a black background. How-

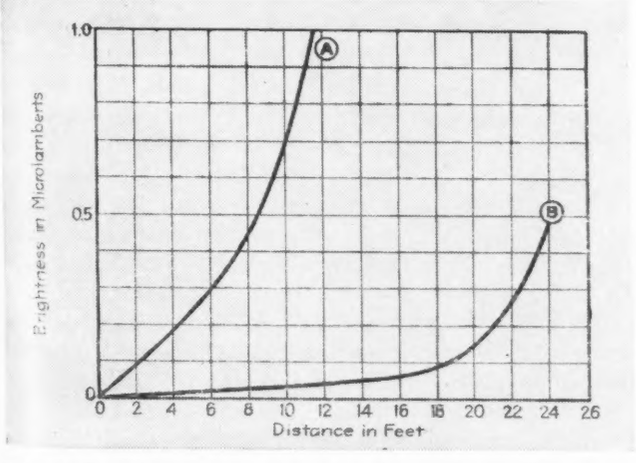

Fig. 6—Reading distance of phosphorescent, yellow-green strontium sulphide. The letters are black, and are viewed against a black background, by the fully "dark adapted" eye.

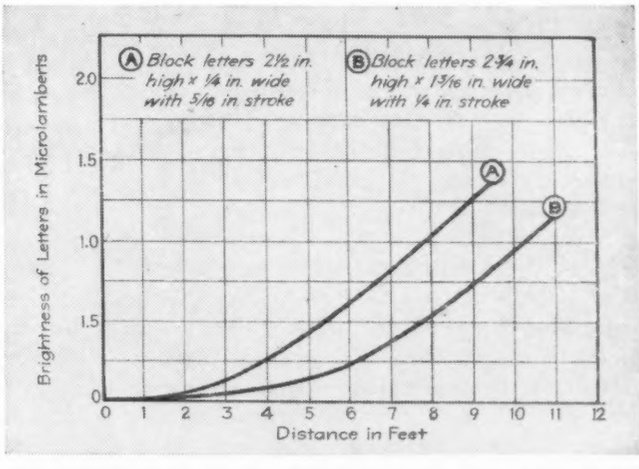
Brightness of Letters in Microlamberts
A Block letters 2½ in. high × ¼ in. wide with 5/16 in. stroke
B Block letters 2¾ in. high × 13/16 in. wide with ¼ in. stroke
2.0
1.5
1.0
1.5
0
A
B
0 1 2 3 4 5 6 7 8 9 10 11 12
Distance in Feet

Fig. 7—Distance at which phosphorescent strontium sulphide spots, 1 in. (A) and 6¾ in. dia. (B), can be seen against a black background, below 1.0 microlamberts brightness.

ever, the phosphorescent material, strontium sulphide, radiates in the blue green region of the spectrum. It is in this region that the dark adapted eye is most sensitive. Such luminous spots and letters could be used to locate valves and light switches for a limited period after room illumination ceases.

EXAMPLE 4. Self luminous radium activated deck markers, personnel markers and rope, unlike phosphorescent tape, have no appreciable decrease in brightness except that occurring over a long period of time, from the disintegration of the zinc sulphide by radioactive bombardment. It has been found that for the standard Navy deck marker, 1 inch in diameter, the most useful brightness is 5 microlamberts. The personnel marker is usually about 10 microlamberts, and rope may vary from 0.5 to 1.0 microlamberts. The advantage of this type of luminous material lies in the property of constantly radiating light, having low intensity but sufficiently bright to be useful. The 1 in. deck marker of 5 microlamberts brightness is visible between 125 and 150 ft. under starlight conditions and somewhat farther in total darkness. The visibility of luminous rope depends on the form into which it is placed.

Legibility of letters and characters made of luminous material involve color considerations. It was stated earlier that fluorescent radiation can be of any color of the visible spectrum; also, a wide range of colored phosphorescent material can be obtained by introducing various metallic impurities in the zinc sulphide phosphor. A practical application of these colored phosphors has been made in the design of instrument dials for airplane use. It has been found that red does not affect the dark adaptation nearly as much as light in the region of the spectrum from yellow through blue.

Airplane instrument dials painted with the usual yellow green luminous zinc sulphide affects materially the dark adaptation of the pilot. To eliminate this difficulty, material suitable for instrument dial illumination was developed. It was, however, much more difficult to find a red phosphor which would withstand radioactive bombardment. An examination of the sensitivity curve of the eye shows that the eye in the dark-adapted state is proportionately less sensitive to red light than the yellow green; thus, to make the letters and numerals on the dials visible, the brightness of the phosphor had to be increased.

Weathering

In the selection of luminous materials it is necessary to examine its ability to resist deterioration from heat, light, and moisture and sometimes acid fumes or mechanical abrasion.

Strontium sulphide is easily affected by moisture, and it has been found both difficult and expensive to provide suitable protection. Zinc sulphide, while quite stable under the action of heat, light and moisture has too rapid a light decay to be useful for most purposes. Calcium sulphide has been found to be the most useful of the three phosphors. While its luminous characteristics leave something to be desired it resists deterioration sufficiently well to serve as aids to night activities. However, with radioactive self luminous aids, zinc sulphide has been found to be the only suitable phosphor, and since it is used in relatively small luminous areas, the problem of protection is less troublesome.

Weathering is also a problem in fluorescent applications. These, however, are more restricted and of such a nature that weathering is not so important. Fluorescent material for airplane instrument dials is practically limited to zinc sulphide, so that the problem of protection from the effects of heat, light and moisture is not too difficult. In laboratory accelerated weathering chambers, where the deterioration caused by heat, light and moisture is more rapid than under service conditions, radioactive markers have lasted over periods which indicate a life at sea of at least 2 years.

Uses of Luminous Materials

Phosphorescent tape and phosphorescent paint can be used as emergency lighting over a short period of time. Their use is limited to places where the material can be activated for at least 5 to 20 minutes previous to use and to places reasonably well protected from exposure to excessive heat, light or moisture. The useful luminous life may be a matter of many hours; on a clear starlit night, in open spaces, white paint will be brighter than phosphorescent material an hour after activation. Table III contrasts the brightness of some form of luminous material with two white surfaces. Starlight illumination is approximately 0.0002 foot candles. The contrast ratio is a measure of the visibility and should not fall below a value of 10, in order that markers and signs can be seen quickly during emergencies.

Adhesive backed tape and paint become dirty after a period of use and cannot be cleaned and must be renewed to be most effective. For many purposes luminous plastic material has many advantages since it can be cleaned with soap and water. It lacks the convenience of the pressure sensitive adhesive backing and cannot be cut to any desired shape as readily as tape. A particularly useful application of the luminous paint or plastic is in marking flashlight cases.

The radioactive luminous personnel or deck marker used by the Navy is an example of a useful application of luminous material. Requiring a light source of low intensity which could be seen not over 150 feet and would continue to function indefinitely without complicated electrical or mechanical arrangements, the Navy tried radioactive self luminous zinc sulphide, with consequent development of the personnel and deck marker and luminous rope. The marker consists of a spot of radioactive luminous material one inch in diameter inclosed in a clear plastic shell, reinforced by a suitable metal ring. The brightness of the luminous surface of the personnel market is approximately 10 microlamberts and the deck marker 5 microlamberts. The brightness decreases slowly with age but the useful life is probably from 2 to 4 years. Increasing the luminous area by placing markers close together would increase the distance at which they are visible; however, it has been found that if the markers are separated a distance of five feet the luminous area cannot be seen beyond the visibility distance of a single market.

Radioactive plastic tubing finds its main uses in marking cargo, outlining landing craft and accessories, and constructing guide lines. Each unit is a five foot length of 3/16 in. clear

Table III—Brightness and Contrast Ratio Under Starlight of White Surface and Some Samples of Luminous Material

Hours After Darkness	1		3		6		9	
Material	Brightness micro-lamberts	Contrast Ratio	Brightness micro-lamberts	Contrast Ratio	Brightness micro-lamberts	Contrast Ratio	Brightness micro-lamberts	Contrast Ratio
White Paint (65 percent)	0.15	6.5	0.13	6.5	0.13	6.5	0.13	6.5
White Tape (90 percent)	0.18	9.0	0.18	9.0	p.18	9.0	0.18	9.0
Strontium Phosphorescent Tape	0.5	25	0.17	8.5	0.10	5.0	0.06	3.0
Strontium Phosphorescent Tape	0.6	30	0.27	13.5	0.20	10.0	0.16	8.0
Radioactive Tubing	1.0	50	1.0	50	1.0	50	1.0	50
Radioactive Marker (1 in. dia.)	8.0	400	8.0	400	8.0	400	8.0	400

Table calculated by Mr. William Frederick when with BuShips.
Contrast ratio is that of the brightness of material to that of a dark grey background illuminated by starlight.

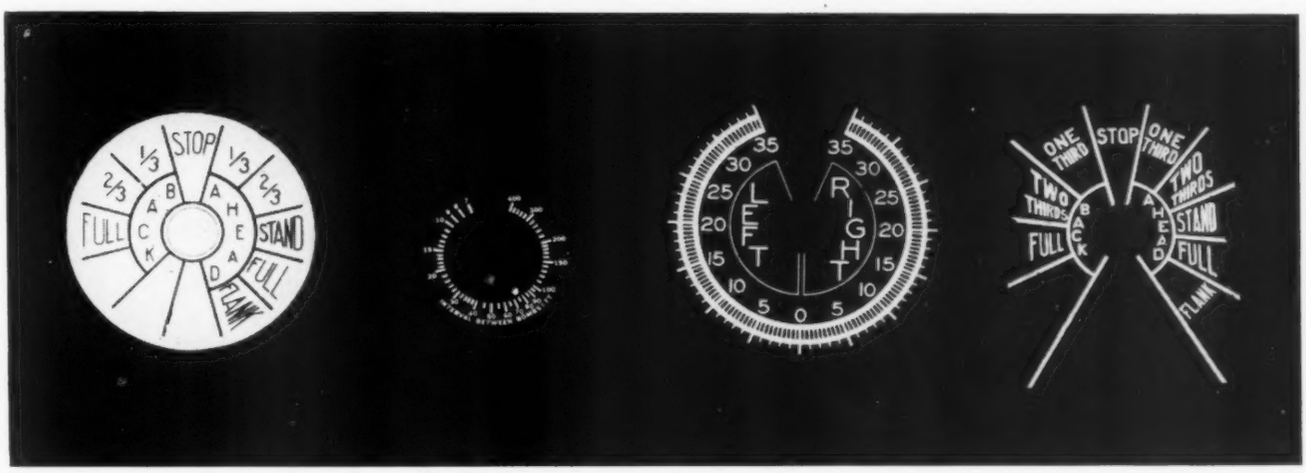

Fig. 8—Navigational instrument dials: Engine room indicator dial markings treated with phosphorescent strontium sulphide; fluorescent bomb sight dial; rudder angle and engine room indicators treated with radioactive zinc sulphide paint.

plastic tubing with clips at each end in order to make continuous lines of any desired length. The inside wall of the tubing is coated with a layer of radioactive zinc sulphide. Brightness of the tubing will average approximately one microlambert. The tubing can stand considerable exposure, with little deterioration. The Bureau of Medicine and Surgery has prepared recommendations for safe handling of these radioactive materials.

Phosphorescent tape is another useful application of luminescent material to aid movements in the dark. This tape consists of a layer of phosphorescent material, from 1/64 to 1/32 in. in thickness, applied to a fabric tape usually 4 in. wide and protected by a suitable, flexible transparent lacquer. On the back of the tape is applied a pressure sensitive adhesive; thus, any desired length of tape can be cut off and applied. This phosphorescent material becomes activated when exposed to light.

Radioactive self luminous materials are used to illuminate airplane and ship instrument and gauge dials, compass cards, and similar instruments. They are also used for "standby" lighting, but can be used where other types of illumination introduce complications.

Application to dials is accomplished by mixing dry radioactive zinc sulphide powder with a suitable adhesive and applying to letters or numerals. The most common color is yellow green, but there are uses where this may affect dark adaptation, such as illumination of the instrument panel of night flying aircraft. For this purpose a red radioactive self luminous material has been developed.

Use of fluorescent material at sea is limited since activation must continue throughout the period of ultraviolet radiation. To get a successful ultraviolet lamp, a mercury or argon arc must be used. While excellent sources are obtainable, they are not rugged or reliable enough in operation to be satisfactory for shipboard use. The same is true of aircraft applications.

Phosphorescent and fluorescent materials of today are superior to those produced at the beginning of the war. They are brighter, longer lived and deteriorate less rapidly under heat, light and moisture. However, they still leave much to be desired. Research in the application of luminous material and in the physics of the phenomena of phosphorescence and fluorescence is extensive. This research can take two directions. One, investigation of the physics of luminescence, the other, attempts to improve by empirical means present luminous substances and to discover others. Because of the complexity of phosphorescence and fluorescence, relatively little improvement on the present material will result from the basic study for a long time, since practice is so far ahead of theory.

ZERO-BACKLASH ARRANGEMENTS

For Gear Trains and Drives

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Methods of achieving smooth action without excessive backlash in spur, bevel and worm gear trains. The schemes described for taking up backlash include shortening the center to center distance, spring loading, eccentrically mounted worms, and spring-loaded pivoted blocks.

A SMALL amount of backlash is not always detrimental in mechanical devices. For instance, it is needed when running gears at high speeds since it provides increased space for the meshing teeth. But excessive gear backlash can cause many mechanical devices to operate erratically and, in extreme cases, prevent operation completely.

Most geared drives operate in one direction for relatively long periods of time and impact loads are infrequent. The gears in an automobile transmission, for example, are often subject to reversals; but each gear makes many revolutions in a given direction before being reversed. In such drives, the load on the gears is removed before the gears start turning in the opposite direction, and a clutch arrangement gradually re-applies the load with a minimum amount of impact as the gears start to revolve. However, in certain mechanical devices the gears are directly connected to the load with no clutch and the action is repeatedly reversed. If gears with appreciable backlash are used in devices such as these, there is a decided impact when the gears meet the load in the reverse direction.

Backlash will also cause "lost motion" or "slack" in a gear train. This condition will seriously affect the operation of mechanisms such as astronomical telescopes, range finders, gun controls and other devices where accurate operation is of prime importance.

With gears having involute profiles, backlash can be reduced by bringing the gears into tighter mesh. Shortening the center to center distance by a few thousandths of an inch accomplishes this, the amount of shortening depending on the degree of inaccuracy in the two gears. When the center to center distance is shortened, "hard spots" may develop in the action in which there is slight tendency for the gears to bind. Continued meshing, however, wears these spots down and ultimate smoothness is frequently possible. It is expensive, awkward and often impractical to have swing blocks and pivoted

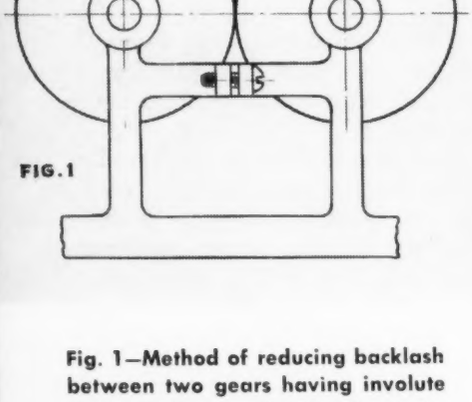


FIG. 1

Fig. 1—Method of reducing backlash between two gears having involute tooth profiles by adjusting the center to center distance between the gears.

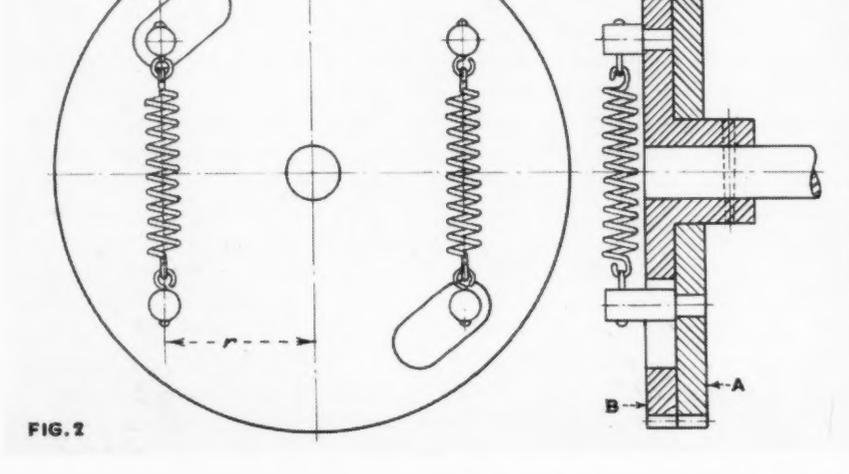


FIG. 2

Fig. 2—Backlash removed in gears by spring-loading. Gear *A* rides on the hub of gear *B* that is keyed to the shaft. By turning one gear with respect to the other, before they are brought into mesh with the pinion, the desired tension is obtained in the springs. The torque produced by the springs must equal the torque transmitted by the gear.

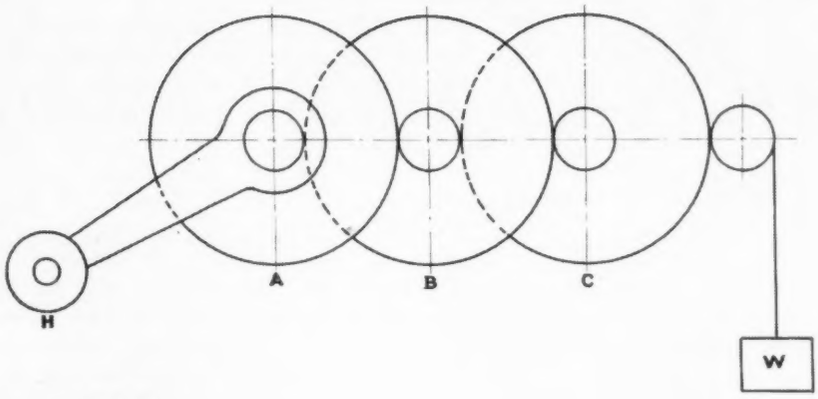


FIG. 3

Fig. 3—Device illustrating removal of backlash by spring-loading. Hand crank *H* raises or lowers weight *W* through doubled and spring-loaded spur-toothed gear train *A, B* and *C*.

Fig. 4—A stiff wire or bar used to spring-load gearing. This method has an advantage in that gears can be assembled unloaded and the spring load applied by the adjusting screw after assembly with the pinion gear to obtain needed torque.

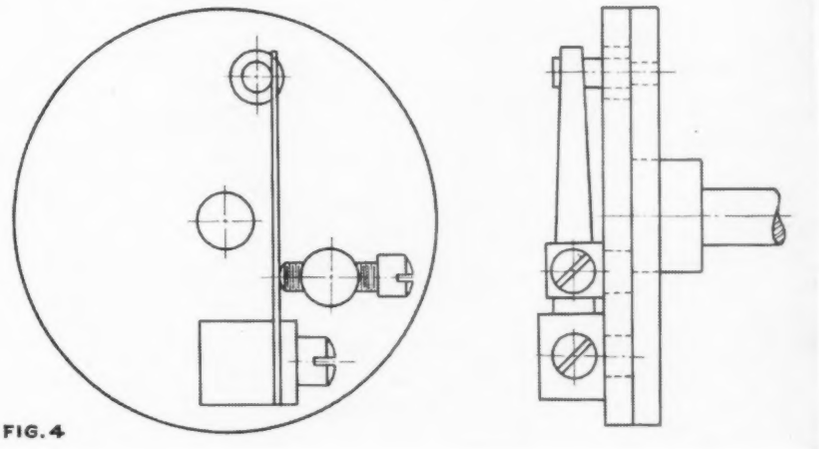


FIG. 4

joints for adjusting center to center distance, but it is sometimes possible to build into the structure enough flexibility as shown in Fig. 1 to permit adequate control.

Most zero-backlash schemes induce more gear wear than will result from ordinary free action. The system shown in Fig. 1 is admittedly bad in this respect. However, before rejecting zero-backlash on this basis, careful consideration should be given to the relative importance of wear. Many devices, similar to that shown in Fig. 1, will work well for long periods of time and under conditions similarly imposed. The scheme is not recommended for high speed gears.

Spring-Loaded Gears

Spring-loading can remove backlash in gears. Although comparatively easy on wear because of increased flexibility, the expense of doubling the gears adds to the cost. An example of spring-loaded gears is shown in Fig. 2. Gear *A* rides on the hub of gear *B* that is keyed to the shaft, and coil springs are attached to each gear as shown. By turning one gear with respect to the other, before they are brought into mesh with the pinion, the desired tension is obtained in the springs. To effectively remove backlash, the torque produced by the springs must equal the torque transmitted by the gear. This is shown by the device in Fig. 3, where a hand crank *H* raises or lowers a weight *W*. In this particular instance, it is required that the action be smooth, even on the change of direction of motion. To accomplish this, it is necessary that each of the three gears *A, B,* and *C,* be doubled and spring loaded. The torques are different in each shaft and greatest in the shaft for gear *A*. If each gear has 92 teeth, each pinion 23 teeth, the weight *W* is one-half pound, and the diameter of the drum on which the weight cord winds is one-half inch, then the torque in shaft *A* is

$$T_A = \tfrac{1}{2} \times \tfrac{1}{4} \times \frac{92}{23} \times \frac{92}{23} \times \frac{92}{23} = 8 \text{ lb. in.}$$

If two springs in gear *A* are placed at an effective radius of 3/4 in., each must exert at least 5 1/3 lb. This does not provide for an acceleration of the mass or losses in friction. It is sometimes more practical to work from the other end. For example, it might be known what the maximum force exerted on the handle would be. From this, the torque in shaft *A* can be found and the required spring tension determined. The tension in the other springs can be smaller in proportion to the torque ratios. In order to bring about enough torque, it may be necessary to use three or four springs in each gear set.

Spring requirements for gear backlash devices are considerably different from those in other devices. To compensate for backlash, the spring moves the gear only a few thousands of an inch. Thus, a coil spring may have to be stretched a quarter of an inch or more to obtain the tension force required, but after the spring and gears are assembled with this tension in the spring, the actual movement or extension of the spring is extremely small. The spring movement is only the amount of the actual backlash. It is therefore possible to use springs that would not be satisfactory for other purposes, since the amount of stored energy needed in the springs is small compared to that required in most mechanical devices. A stiff wire or bar can be used, and this need be flexed or energized only a small amount to bring about the required force. This type of spring loading is shown in Fig. 4. It has a distinct advantage in that the gears can be assembled unloaded and the spring load applied by the adjusting

screw after assembly with the pinion gear. This saves much time and effort.

Bevel gears can be made to operate without backlash effect, but the actual arrangement depends on the size of the gears. One method that works well for small bevel gears is shown in Fig. 5. In machining the gears, the two portions of the pinion gear can be clamped together by screws, using the holes that afterwards support the pins. Thus, the teeth on both portions can be milled together in one operation.

A worm and gear can be made to operate with zero backlash by making the center distance adjustable. This is accomplished as shown in Fig. 6. The worm gear in this instance must be a helical gear. The two bearings *A* and *B* have the shaft bore eccentric with respect to the outside of the bearings. Two small holes drilled in the ends of the bearings allow them to be turned by a small spanner wrench. By turning these bearings together, the worm can be brought into tight mesh with the gear and smooth action obtained with practically no backlash.

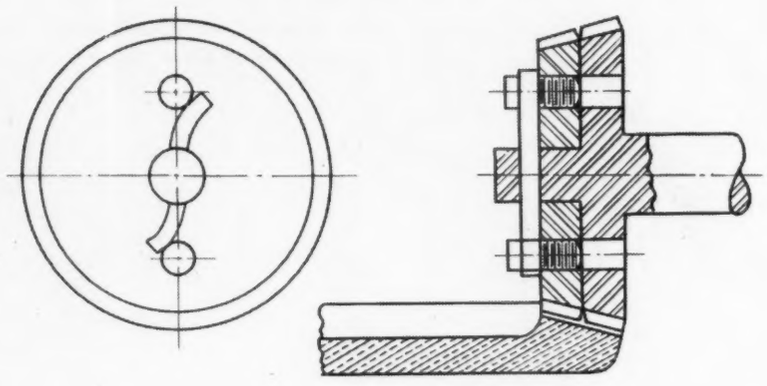


FIG. 5

FIG. 6

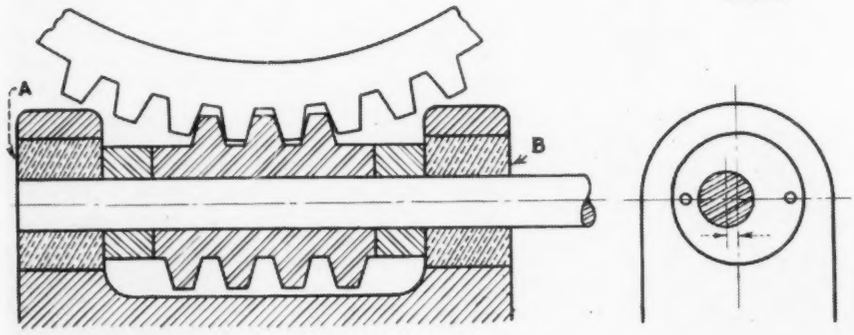


Fig. 5—Method for eliminating backlash in bevel gears. In machining, the two portions of the pinion gear can be clamped together by screws, using the holes that afterwards support the pins. The teeth on both portions can be milled together in one operation.

Fig. 6—Method of eliminating backlash in worm and helical gear. The bearings *A* and *B* have the shaft bore eccentric with respect to the outside of the bearings. Two small holes drilled in the ends of the bearings permit them to be turned easily with a spanner wrench. By turning bearings together, the worm can be brought into tight mesh with the gear.

Pivoted Worm Block

Another method of achieving smooth action without backlash in worm gears is shown in Fig. 7. In this device the worm block is pivoted and a leaf spring presses the worm into close mesh with the gear. This arrangement requires more than normal space, but gives very smooth action and long wear. The bearing block that supports the pivot *A*, however, must be well designed, since there is a large lateral moment at this point that tends to turn the entire block sideways.

All of these schemes are practical, but their satisfactory operation depends upon a high degree of precision machine shop work. This is not because of any shortcomings of the backlash devices. It is hardly practicable to go to the trouble of using backlash devices and then obviate their effect by allowing poor fits and lost motion in other parts of the device.

FIG. 7

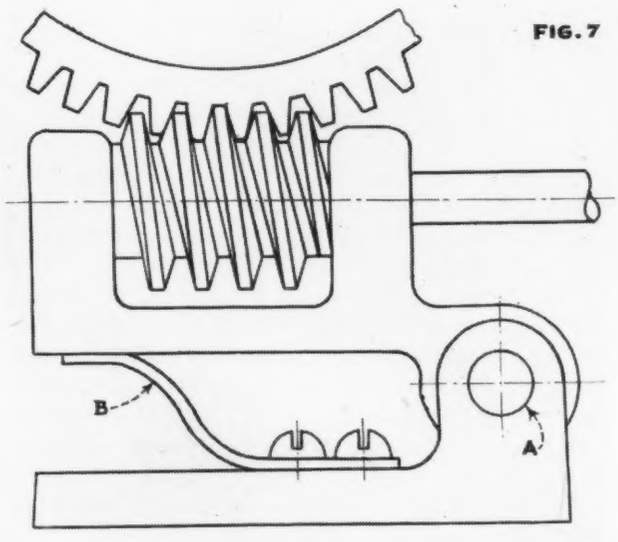


Fig. 7—Elimination of backlash in worm gears. Worm block is pivoted at *A* and leaf spring *B* presses worm into close mesh with gear. This arrangement requires more than normal space, but it gives smooth action.

TECHNICAL NEWS HIGHLIGHTS

Diamond Die Drilling Method Developed by Bureau of Standards

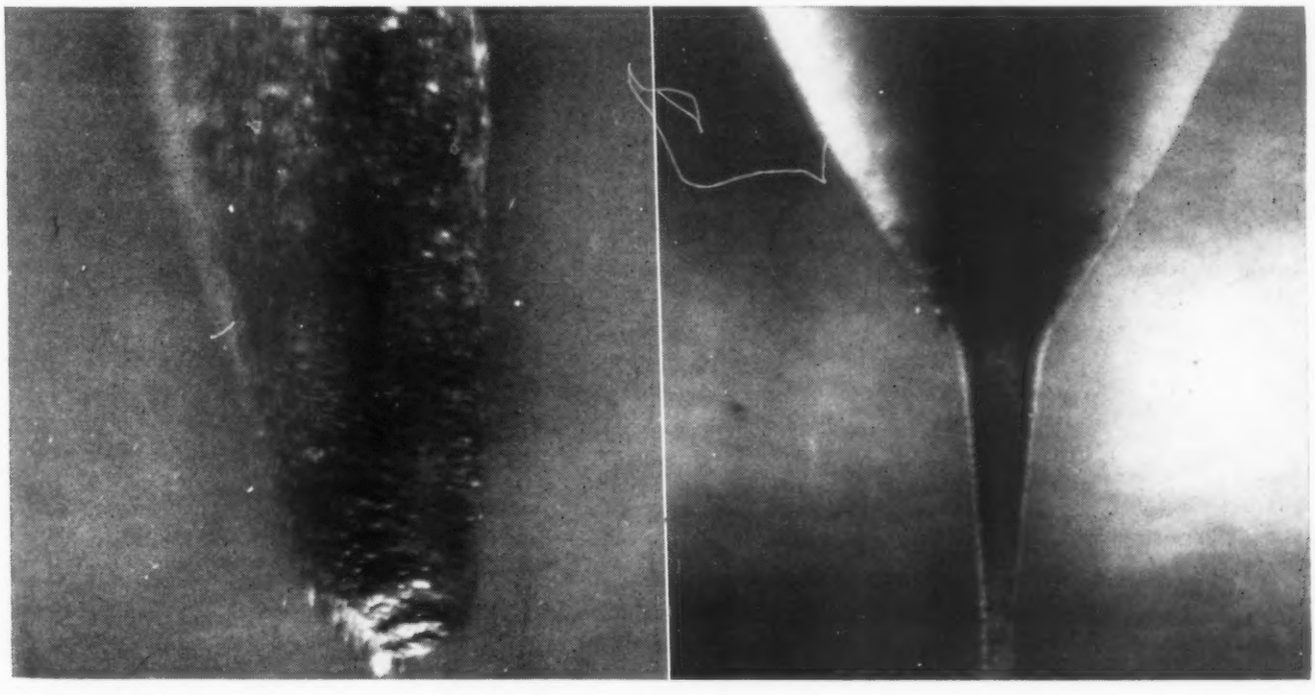
AN ELECTRICAL METHOD, for drilling small diamond dies, used in drawing and shaping extremely hard and fine wire, has been developed at the National Bureau of Standards, eliminating almost 100 man-hours from older processes. This discovery has revolutionized the fine-wire diamond die industry.

Although dies of tungsten, molybdenum, or boron carbide have competed with diamond dies, the diamond is still the only tool used in shaping the harder and tougher materials, such as chrome, nickel, brass and phosphor bronze; and high-carbon steel wires. Also wires finer than 0.015 in. in diameter can be drawn only by diamond dies. The diamond is indispensable whenever precision of size and perfect roundness are essential considerations. Important uses for diamond-drawn wire are in radar and radio equipment, in high-frequency coil windings, as filaments for small tungsten lamps and in delicate and precise electrical instruments. Wires of this type have become increasingly important with the expansion of the electronic and allied industries.

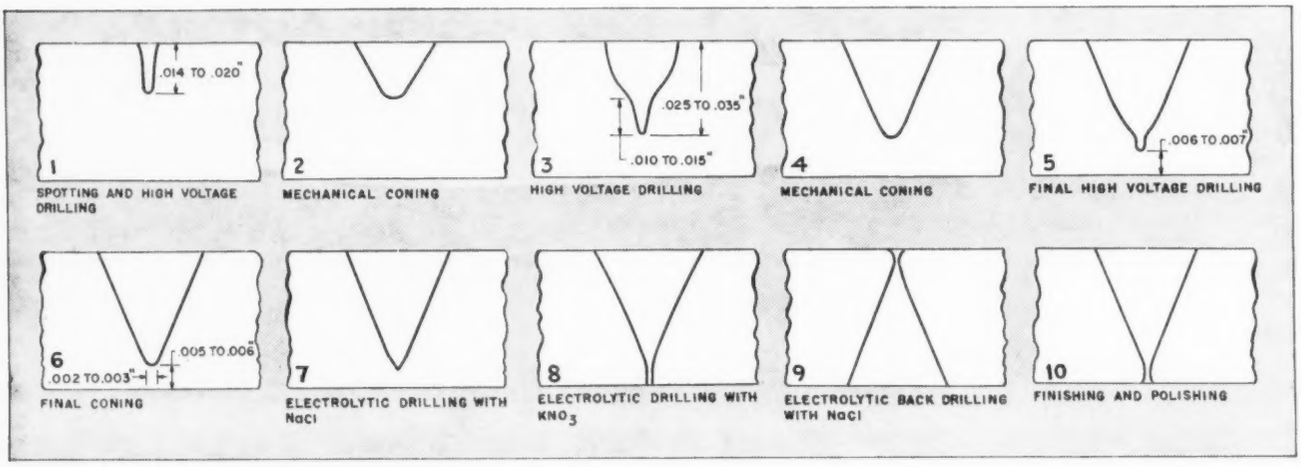
During the war, with all supplies from Europe cut off the National Bureau of Standards undertook an investigation to speed up the manufacturing process and improve the quality of dies to meet the emergency requirements for many vital war products.

A diamond die consists of five surface areas: the primary cone, the secondary cone, the reduction cone, the bearing and the exit cone. In appearance such a die is similar to a wine glass. The portion of the die where the wire enters the primary cone, corresponds to the goblet part of a wine glass. The narrowing stem from the bottom of the goblet to the base of the glass corresponds to the secondary cone, the reduction cone, and the bearing. Actual wire shaping takes place at the bearing. Finally, the base of the wine glass represents the exit cone.

The primary cone is drilled by means of high-voltage sparks formed at the point of a needle electrode in contact with the face of the diamond. These sparks release the energy stored in a condenser which has been charged to a high voltage by a transformer. The



(Left) The primary cone in a diamond die before mechanical coning. It is made by three applications of high-voltage, followed each time by mechanical coning. (Right) Two steps in making the secondary cone at the apex of the primary cone. The first requires 30 minutes of electrolytic drilling with a 5 percent solution of sodium chloride. The second step involves 40 to 45 minutes of drilling with a solution of potassium nitrate, piercing the back of the diamond die.



Ten successive steps in the National Bureau of Standards method of making small, diamond dies. This method, combining electrical and mechanical operations, has reduced the average time for making a die by more than 100 hours.

condenser is charged through a "quenched" gap consisting of a number of very small gaps in series.

The secondary cone is formed by the action of a low voltage spark in an electrolytic solution. The diamond is mounted on an insulating post in a shallow glass dish and enough of the electrolyte is used to fill the dish and just cover the diamond. The "drilling" electrode is a platinum-iridium needle that is lowered into contact with the bottom of the primary cone and rests with very light pressure (around one fifth of a gram) on the diamond. A second electrode dips into the solution at some distance from the diamond.

When a low voltage (around 90 volts) is applied between the electrodes, sparking occurs at the tip of the drilling needle and a smooth-walled conical hole is formed in the diamond directly under the needle. The shape of the hole and the angle of its walls are controlled by the type of electrolyte used.

Dies produced by this method have better wearing characteristics in service than those produced by purely mechanical methods. In mechanically drilled dies, holes are started in the top and bottom of the diamond by "bruting" or gouging the face of the diamond with a small diamond chip. The hole is drilled through the body of the diamond by "pecking" it with a spinning steel needle charged with diamond powder, a sort of "star-drilling" process. These operations result in severe strains and minut fractures in the walls of the die cones.

Cones produced by either the high-voltage or the electrolytic drilling process show no strain in the surrounding material when examined under polarized light. Freedom from strain and fractures results in less wear as the die is used to draw wires. The electrical methods of die formation are therefore particularly suitable for small dies used in drawing wires, 0.0015 inch in diameter, or smaller. As a result, domestic die makers now find it advantageous to produce these small dies, formerly obtained through import.

Machine Tool Show Draws Wide Interest

THE 1947 MACHINE TOOL SHOW, to be held in Chicago in September, has captured the interest of industrial executives throughout America, as well as the rest of the world, judging from the deluge of requests for advance information being received daily by the National Machine Tool Builders' Association. Herbert H. Pease, President of the Association and President of the New Britain Machine Company, reported that the Association's offices in Cleveland are receiving a steady flow of inquiries about the show. These are not only requests for information, but are actual advance registrations. Registrations are particularly heavy from South America.

Theme of the show is "More Goods for More People at Lower Cost." The show will be held in the Dodge-Chicago plant, the only factory building in the world large enough to hold an exhibit of such magnitude, from September 17 to 26 inclusive. More than 1,000 machine tools and related equipment and appliances will be shown in actual operation. The Dodge-Chicago plant has floors capable of supporting the heaviest loads required, ample power, and all the facilities for the operation of machine tools under working conditions.

Concurrently with the show will be held the Machine Tool Congress, with nightly sessions at which papers and discussions will touch upon the latest developments in the metalworking industry. K. H. Hobbie, President of the Chicago Technical Societies Council, is president of the Machine Tool Congress. Nine important associations will participate in arranging the program of speakers at the meetings, which will be open to registered visitors at the Machine Tool Show.

Included are: American Bolt, Nut and Rivet Manufacturers' Association; American Foundrymen's Association; American Machine Tool Distributors' Association; American Society of Mechanical Engineers; American Society of Tool Engineers; Chicago Technical Societies Council; National Electrical Manufacturers' Association; Society of Automotive Engineers and the National Machine Tool Builder's Association.

Assignment of space in the show has been completed by the show committee headed by Swan E. Bergstrom, Sales Manager, The Cincinnati Milling Machine Company. More than 250 American manufacturers are represented. Advance registration forms should be mailed to the National Machine Tool Builders' Association, 10525 Carnegie Avenue, Cleveland 6, Ohio. Hotel reservations should be made by mail to the Chicago Convention Bureau, 33 North La Salle Street, Chicago 2, Illinois.

WASHINGTON NOTES

Scope of Scientific Work To Be Reported Soon

Reliable figures on the scope of scientific work in this country soon will be available for the first time. A special board, appointed by President Truman last October will bring out a series of reports in August and September. Because of constant changes in personnel, the Scientific Research Board was late in getting started and its reports are not likely to contain any definite recommendations for action.

One report will be an inventory of the nation's scientific and technical manpower weighed against estimates of need and of students being trained. This should provide a check on the much talked of wartime gap in scientific education. Early indications are that the survey will show adequate numbers of top scientists and research assistants but a rather serious deficiency in the intermediate level.

Equally significant will be a report on the volume of expenditures on scientific work—both government and non-government. It is hoped that there will be a breakdown between basic research and developed work. Reliable figures of this type have never been abailable, although frequent estimates have been attempted.

Federal expenditures on science will be broken down in still another report. This has been a mystery since the end of the war. The form of the Federal budget makes it difficult to isolate scientific programs and responsible estimates of Federal research work have ranged all the way from a few hundred million to a billion dollars.

A fourth report will be a "critique" of the Federal research program. This seems likely to be the weakest report of all—although this was the primary purpose of the Board. The President's Scientific Research Board was created last fall under John R. Steelman, mostly at the inspiration of one of Steelman's right hand men in the Office of War Mobilization and Reconversion—one James Newman. The original purpose was to check on widespread accusations that the military were assuming disproportionate controls over science and that there was a great deal of duplication among Army, Navy, the Atomic Energy Commission and non-government groups. Soon after the Board was named Newman left the government to become one of Henry Wallace's right-hand men on the *New Republic*. A new group under Steelman has set up a staff at the White House with J. Donald Kingsley as Executive Secretary. This group hasn't had time to probe too deeply into details of the government program and will probably content itself with a general description of the work of the various agencies.

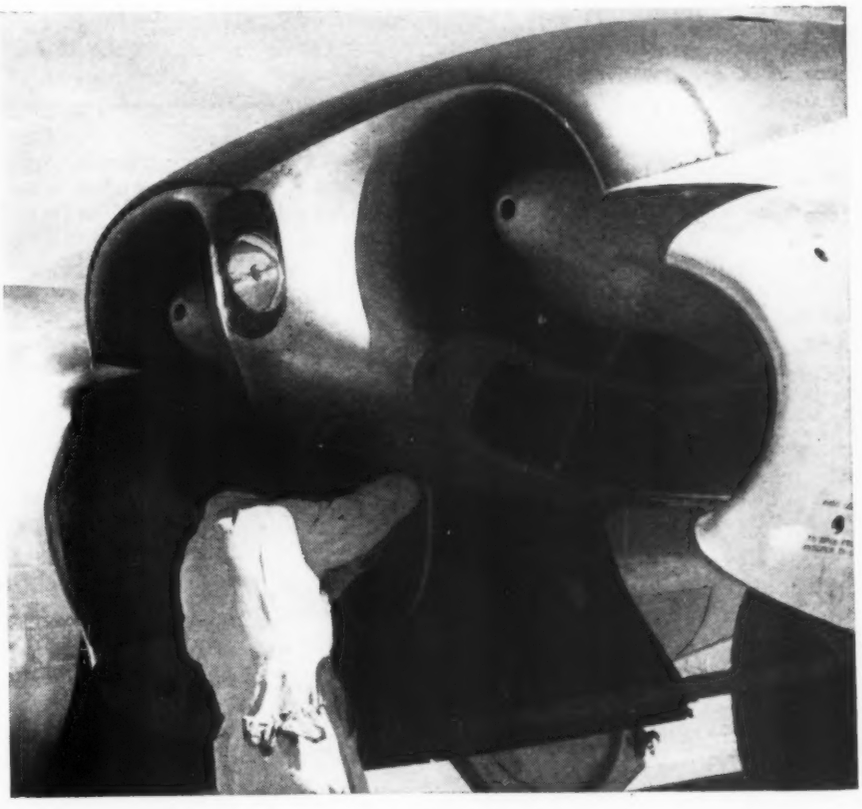
SOME CONCLUSIONS—Industry must greatly increase research and development work during the next five years—perhaps as much as double it. This is the conclusion to which the White House advisers are coming, in working up an Administration program designed to provide for the needed expansion. The White House group is also convinced of the need for greatly expanded non-military research. They're now engaged in working out specific government action. No final decisions have been made, but here are some of the details being considered:

TAX REVISION—Tax laws would be modified to provide tax incentives for industrial expenditures on research and development. This might be tied in with proposals for tax measures designed to encourage modernization and plant expansion to take advantage of new discoveries.

SUBSIDIES—This is to be a future consideration rather than an immediate proposal. But federal financing of non-military research, by industry as well as by universities, may be "kept in mind" as a preferred method of government pump-priming, to be taken used in times of depression.

For the present, it is not money that limits research so much as a shortage of scientific manpower. All of the half-billion dollars availabe this year for miitary research and development will not be spent because trained men are not available to do the work. The military program is putting a strain on industrial scientific manpower and is seriously cutting into teaching staffs.

The forthcoming report will show that in 1942 there were some 40,000 college-level teachers of science, of whom about 15,000 had a doctorate. Today, with student enrollments in sci-

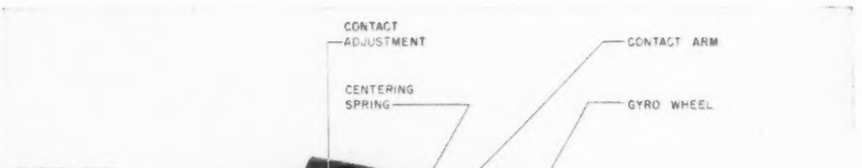


PLACED ENTIRELY AHEAD OF THE WING, engines in the Army Air Forces' first four-jet bomber, North American Aivation's B-45, are accessible from ground for easy maintenance. Each nacelle holds two General Electric J-35 engines, manufactured by Alison Division of General Motors. Controls are easily operated by a hydraulic "boost" system, which provides 95 percent of the energy required to operate the surface controls. AAF has ordered 96 of the bomber, which flies at a speed described as "exceedingly fast."

ence departments practically doubled, the number of teachers has risen only to 50,000 and the number with a doctorate has decreased to about 13,000.

Hardest hit are the small colleges. This is considered especially serious in view of the rather surprising evidence that a larger proportion of small school graduates go on to advanced study and professional scientific work. That is why, in current Senate debates, the administration forces are backing a requirement, in Science Foundation legislation, that at least 25 percent of funds be distributed among the states.

With manpower a critical factor, another major recommendation of the forthcoming reports will be built around stricter regulation of federal research. It is recognized that some agency is needed to see that federal departments do not waste scarce skills on non-essential or over-lapping projects. Last year it was expected that the proposed National Science Foundation might do this job. But it is now thought that the foundation, if it is set up at all, will be controlled by a part-time board not geared for a federal coordinating job.



National Bureau of Standards

CUTAWAY VIEW OF GYROSCOPIC STABILIZER used in the Bat, a wartime missile which automatically "homed" on a selected target, guided by radar echoes of the pulsed microwave radiation emitted by the missile. Designed to glide silently at the speed of a fighter plane, with a 1000-pound bomb as cargo, the Bat was carried under the wing or fuselage of a Navy fighter or bomber and was released at a distance as great as 14 miles from the target. Once the crew of the mother plane located a ship or other target, the missile was set for this particular target and the release button pushed. From then on the Bat automatically followed every maneuver of the target until it eventually struck home, with the mother plane free to proceed as it pilot wished.

HOW MUCH RESEARCH? Both military and civilian research expenditures are running far ahead of prewar, the White House study shows. However, non-military research represents about the same percentage of national income as before the war. Total government and private expenditures, exclusive of the Atomic Energy Commission, will total about 1.1 billion dollars this year. Atomic research might add another 100 million dollars. Of this total, a third is for basic research. The remainder is for specific development work. Nearly half of this expenditure, some 450 million dollars, is for government-financed military work. Money available for military research this year is nearer 500 million dollars but not all of it will be spent.

Non-military expenditures will be apportioned as follows:

$100,000,000 . . . federal
$450,000,000 . . . industrial
$100,000,000 . . . university and research institutes

Looking ahead five years or so, the White House advisers assume that military research should be continued at something like the present half-billion dollar a year rate. They think there are strong economic reasons why non-military expenditures will have to expand greatly. Even though industrial and university research is now much greater than before the war, the study is expected to point to major changes in the technological situation which render present activity inadequate:

We can no longer "import" basic reseach as we did before the war. During the war, basic research practically stopped. It will be years before the shattered laboratories and universities of Europe are reestablished. If we want a stock of increasing knowledge about the nature of the physical world we will have to build it ourselves.

We are in danger of falling behind the rest of the world technologically. One reason for U. S. industrial leadership in the twentieth century has been that we were industrial "late-comers." Like Germany we had a new industrial plant to compete with a world bogged down by vested interests in obsolescence. But capital investment nearly stopped in the thirties, and was directed to the specialized needs of war in the early forties. When Europe rebuilds its industries, it will have the modern equipment.

It is felt that a high level of industrial investment is the real answer to this. But that calls for a correspondingly high level of research—both to guarantee that new facilities are truly up-to-date and to provide economic incentive to replace obsolete plants.

Industry faces major technological changes in order to deal with depletion of such major resources as petroleum, many non-ferrous metals, and good iron ores. It must prepare for many shifts into alternate materials.

It is this sort of reasoning which will underly recommendations that government policy should shoot at a level of basic research perhaps four times present activity and industrial research and development something like double what it is now.

Patent Report Released

On one subject—patents—the White House reports will be conspicuously silent. Presumably this reflects a realistice conclusion from the fate of last year's Kilgore bill, that resistance to any meddling with the present patent set-up is too strong to buck. But officially, the reason is that a Justice Department report to be released soon defines administration policy.

Theme of this report is the handling of patents on discoveries made in the course of government research contracts with industry. The study was begun in 1943 and was completed last fall. It

includes a vast bulk of purely factual material on how wartime research was handled—plus some highly controversial recommendations for the future.

As drawn up by the Justice Department, the report recommended that all government-financed discoveries be patented and the patents held by a federal patent custodian. The custodian would issue royalty-free non-exclusive licenses on a selective basis; he would deny them where "monopoly" might be promoted, issue them where competition would be served. At one stage, it was recommended that temporary exclusive licenses might be issued.

These recommendations were bitterly opposed by the armed services. They feel that good relations with industry demand continuation of the present practice—letting the firm that does the work have the patent subject only to a royalty-free license to the government. Military objections held up the report at the White House until now—with the controversial features being alternately inserted and withdrawn. As finally approved by the President, the report is understood to be in substantially the form in which it was written by the Justice Department last fall.

Science Foundation Dead

It was similar reasoning to the above, although with less factual data behind it, that underlay the agitation of the past two years for establishment of a National Science Foundation. But the foundation still looks like a project for the future.

The bill which the Senate passed two months ago (see June "Washington Notes") is much less controversial than the Kilgore measure which passed the Senate a year ago but was shelved in the House. It by-passes the controversial patent question and it makes concessions to those who fear federal domination, by splitting authority between a part-time board and a presidentially-appointed administrator. Its supporters have scaled down their ideas to about a $40,000,000 a year level.

But even so, the bill has practically no chance of passing the House at this session and little chance of becoming law next year. There just is not enough pressure behind it. The foundation scheme will undoubtedly come up again—and with a lot more appeal—a few years from now when the veterans educational benefits run out.

High-Temperature Porcelains Take Sustained Load

ALL-CRYSTALLINE PORCELAINS, having exceptional mechanical and dielectric strength at temperatures up to 2000 deg. F., have been developed in the Porcelain and Pottery Laboratory at the National Bureau of Standards. Such porcelains are particularly valuable for sustaining loads at elevated temperatures, as electrical insulators at elevated temperatures, and as insulators for such special applications as aviation spark plugs and radar transformers.

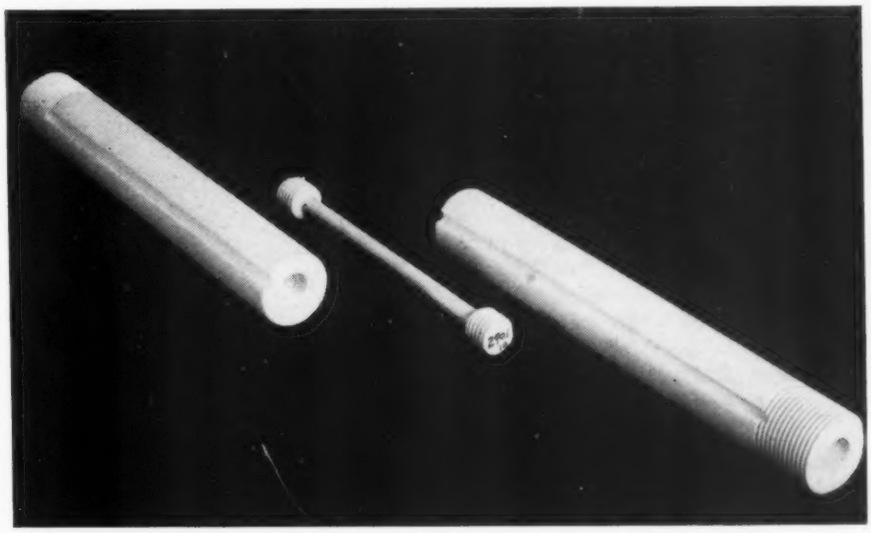
The composition of these new porcelains is characterized by the use of such materials as alumina, beryllia, zirconia, and thoria, with minor additions of other metallic oxides, and by the absence of silica. A representative high-beryllia porcelain contains 84 percent beryllia and 8 percent zirconia, with small amounts of lime and alumina; a typical high-zirconia composition is 80 percent zirconia, 10 percent beryllia, and 10 percent magnesia.

In 1940 the Bureau began a series of investigations to develop glass-free ceramic bodies of the porcelain type which would have simple compositions favorable to high mechanical strength, particularly at elevated temperatures, and good resistance to thermal shock, properties not found in the conventional porcelains.

The work was accelerated by wartime demands for a material that would resist deformation at temperatures up to 2000 deg. F. for use in jet propulsion and gas turbine engines. An advantage of ceramic materials for this purpose, if other properties are satisfactory, is their low density as compared with the high-temperature metallic alloys. Accordingly, the investigation was extended under the sponsorship of the NACA to include strength and creep characteristics in pure tension at temperatures ranging from 1500 to 2000 deg. F. On the basis of preliminary tests, six compositions were chosen for further study.

Tests show that all of the compositions are highly resistant to deformation or creep at temperatures up to 1800 deg. F for long periods of time. A typical specimen showed an elongation of 0.23 percent after 2600 hours, during which the applied stresses ranged from 9,000 to 14,000 lb. per sq. in. At temperatures of 1900 deg. F. and higher, however, the tendency to creep is more pronounced and appreciably lower stresses produce a creep rate measurable by the hour.

Research and development are continuing in the field of high-temperature porcelains, for ceramics offer definite promise of usefulness in the high-temperature regions where metallic alloys are no longer serviceable.



A porcelain specimen (center) at the National Bureau of Standards for tension tests at elevated temperatures and ceramic adapters, threaded on both ends, through which stress is applied. The two adapters screw into brass heads which connect through universal joints to the loading device.

Foreign Reports

. . . . . from McGraw-Hill World News

New British Polymer Resembles Nylon

Full-scale production of fiber and fabrics made of "Terylene," polyester made from ethylene gylcol and terephthalic acid, has been started by the plastics division of Imperial Chemical Industries, Ltd. Equipment for melt-spinning of the polymer into fiber is already in operation in I.C.I.'s plastics division at Welwyn. In this process, invented by du Pont for the spinning of nylon, no solvent is used. The polymer is supplied to the spinning head in chip form. There it is heated to a temperature above 246 deg. C. until sufficiently fluid to be pumped through a filter and a thick spinneret plate. The latter is perforated with the correct number of holes to give the desired filament count in the finished yarn.

I.C.I. plans to produce Terylene not only as a continuous filament low-twist yarn analogous to silk, but also as heavy industrial cords. By cutting the filaments into short lengths, staple fiber can also be produced, from which fabrics similar in appearance to cotton or wool can be processed on ordinary textile machinery. Because of its high resilience, Terylene can be expected to give a warm "lofty" staple fabric.

Terylene resists light and heat much more than any organic fiber in current use, and also has a high resistance to moisture. It has a high modulus of elasticity which should make it easy to process. Extremely fine filaments, below one denier per filament, have been produced. The material is also resistant to creep under load. Like other synthetic fibers, it is highly resistant to chemicals and fungus.

Because of other properties, including its electrical-insulating qualities, Terylene is expected to meet with a large demand for industrial purposes, where dyeing is not required. Fibers of high tensile strength, as much as 50 tons per sq. in. can be produced.

Reciprocal Patent Legislation In New Zealand and Australia

The Boykin Act, passed by the United States Congress and promulgated Aug. 8, 1946, which is effective from the same date this year, has already brought legislative response from New Zealand, and Australia is considering similar action. The act extends temporarily the time limit for filing patent applications by residents of countries which before Aug. 8, 1948, extend reciprocal privileges to the United States. It is likely that an analogous bill, to become effective before Aug. 8, 1948, will be introduced in Australia.

Russian Stainless Steel Contains Nitrogen

The possibility of partial substitution of nitrogen for nickel in the production of stainless and heat-resistant steel has been ascertained by Professor A. M. Samarin of the Sicentific-Research Metallurgical Institute of the Soviet Academy of Science. He has developed a new brand of chrome-nickel steel containing nitrogen. Regarding its heat-resisting properties, he claims that steel containing from 22 to 25 per cent of chrome, 13 per cent of nickel and 0.25 per cent of nitrogen is not inferior to ordinary heat-proof steel containing 25 per cent of chrome and 20 per cent of nickel.

Standards Institution Started in India

The General Council of the Indian Standards Institution, set up by the Government of India in September held its inaugural meeting in New Delhi on January 6, 1947. The Council, which consists of 64 representatives from the Central Government departments, Provinces, States, Research Institutions and Chambers of Commerce, approved the creation of the Indian Standards Institution, a non-official body for evolving National Standards.

The Institution will have five sections for textiles, engineering, building, chemical and food and agricultural products, each controlled by a Divisional Council. The actual preparation of standards will be done by small committees of experts representing the various interests such as the producers, consumers and technical experts. The intention, at present, is that the Institution should start actively with engineering and textile products, and as more experience is gathered and the work of the Institution receives the confidence of industrialists and consumers, to extend the activities to other divisions.

A $1,200,000 National Physical Laboratory to be built at New Delhi shortly by the Central Government as part of a wide scientific development program is scheduled to work in close collaboration with the newly established Indian Standards Institution in developing better industrial standardization in this country.

At present there is no well-equipped laboratory in India to undertake work on standards or inter-comparison of standards, so the nine divisions of the new Physical Laboratory are expected to be given many important assignments. These divisions are: Weights and Measures, Applied Mechanics and Materials, Heat and Power, Optics, Electricity, Electronics and Sound, Building and Housing Research, Hydraulics, and Analytical Chemistry.

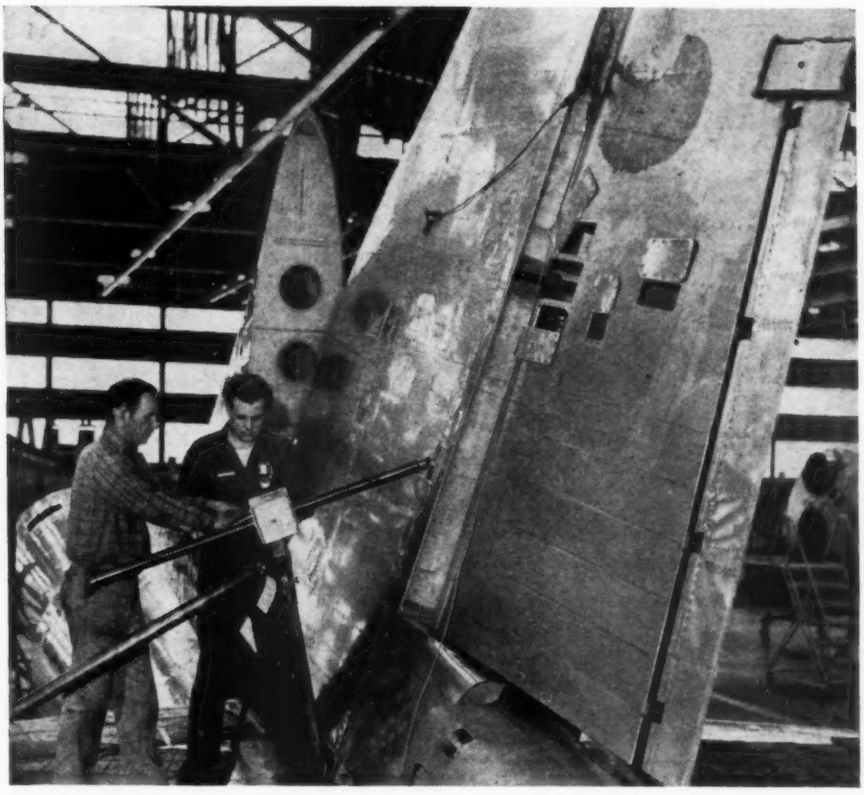
The site chosen for the Laboratory adjoins the Indian Agricultural Research Institute and it is planned that the two units will share a technical library which will form the nucleus of a central information service for Indian scientists.

Australian Industrial Research

Australian industry will be able to have research work carried out at the Commonwealth Research Laboratories at Marribyrnong (Vic.) and Lidcombe (N.S.W.). The laboratories were expanded during the war at a cost of $20.000.000. As the Government will not have sufficient work for the joint staffs of 700, their facilities in future will be available for industrial research.

Indian Patent System

The urgent need for a complete overhauling of the Indian patent system was voiced recently by K. Rama Pai, Government of India Controller of Patents and Designs. Speaking at a meeting of industrialists and engineers he said that a group of industrialists, Government officials, economists and engineers should be named as soon as possible so as to tune the patent laws to India's industrial expansion. Pointing to the need for greater encouragement of Indian inventors he cited the fact that of 1,000 patents applied for in 1925 only 71 were sought by Indians, and that in 1946 there were only to 2,610 applications, of which 266 were by Indians.



FOLDING TAIL which enables new Boeing Stratocruisers, C97A stratofreighters and B-50 superbombers to be housed in standard airline hangars has been undergoing structural and aerodynamic proof tests at Boeing Aircraft Co., Seattle. Top of vertical tail is 38 ft. 3 in. from ground, yet by use of hinge system, overall height can be reduced to 26 ft. 7 in. Here mechanics are shown lowering vertical tail to horizontal position atop right stabilizer. Tripod fits onto left stabilizer, with double screw arm fitting onto fin itself. After removing four belts and substituting hinge pins in auxiliary fittings, vertical tail can be lowered to clear hangar roofs or for easier servicing of rudder and contols.

Technical Societies Council Organized in New York

THE TECHNICAL SOCIETIES COUNCIL of New York has been incorporated recently under the laws of the State of New York. A War Production Committee was organized during the early days of the war at the request of Donald M. Nelson, and subsequently came under the direction of J. A. Krug. It was disbanded following the war after serving many useful purposes. This was the nucleus which formed the Technical Societies Council of New York. The Council is composed of engineering and technical societies in Greater New York and has as its main objectives the advancement of engineering, scientific or technical knowledge and the practice of high professional standards.

The following 14 societies are charter members of the Council and it is expected that other engineering, scientific and technical societies, whose qualifications conform with the requirements of the Council's Constitution, will become members:

American Institute of Chemical Engineers, American Institute of Electrical Engineers, American Institute of Mining & Metallurgical Engineers, American Society of Heating and Ventilating Engineers, The American Society of Mechanical Engineers, American Society for Metals, American Society of Safety Engineers, American Society for Testing Materials, American Society of Tool Engineers, American Chemical Society, American Welding Society, Illuminating Engineering Society, Institute of Radio Engineers, Society of Motion Picture Engineers. The following men are on the Board of Directors: H. C. R. Carlson, President, ASME, H. P. Wall, Vice President, ASSE, M. P. Davis, Secretary, ASTM, H. F. Dart, Treasurer, Institute of Radio Engineers.

Structural Crack Tests Developed by Air Forces

THE RESISTANCE OF AIRCRAFT STRUCTURAL MATERIALS to spontaneous cracking while exposed to the atmosphere under stress can be estimated more quickly on the basis of two new criteria developed by the U. S. Army Air Forces according to a research report on stress corrosion resistance, now on sale by the Office of Technical Services, Department of Commerce.

The report was prepared by Major Baxter C. Madden, Air Corps, for the Air Materiel Command, Wright Field, Dayton, Ohio. Part I is an early progress report dealing mainly with the methods used to expose and stress three types of test specimens. Part II describes the development and application of new standards for measuring stress corrosion resistance, and lists 23 aluminum alloys and 12 magnesium alloys in order of decreasing resistance to corrosion. Certain of the alloys tested had shown definite tendencies to crack under stress when used in aircraft structures while other appeared to be resistant.

One of the new stress corrosion resistance criteria involves strength measurement after accelerated exposure and stress and the other, elongation measurements under similar conditions. Strength sensitivity is considered the better of the two but elongation sensitivity may be used to check strength sensitivity findings.

Stress corrosion resistance was investigated to obtain improved test methods. Several types of accelerated exposure tests simulating aircraft service conditions were tried. The test considered most satisfactory for magnesium alloys was intermittent immersion of static tensile specimens in distilled water, approximating the corrosive effect of vapor condensation either at high altitudes or on the ground, as well as of rain.

In the aluminum alloys intermittent immersion of static tensile specimens in a 3½ percent sodium chloride solution was found to approximate the corrosive effect of sea spray. A large number of wetting and drying cycles accelerated the corrosive action of both aluminum and magnesium alloys. Twice-daily wetting of static cantilver specimens proved unsatisfactory for accelerated testing.

Tests of corrosion under stress were conducted with static tensile, static cantilever and fatigue cantilever specimens. A marked reduction in the exposure time was obtained by use of an exposure stress of 90 percent yield strength with the static tensile type of

specimen, instead of a stress of 75 percent yield strength. The report suggests that 90 percent yield strength as an exposure stress may also reflect actual service conditions, including residual stresses due to severe forming operations, heat-treating, welding, poor fitting during assembly or combinations.

New Developments Feature Metal Powder Meeting

AN EXHIBIT OF METAL POWDERS, products and equipment supplemented the technical sessions of the third annual spring meeting of the Metal Powder Association, held on May 27 at the Waldorf-Astoria Hotel in New York. Widespread interest in the metal powder meeting was indicated by the extremely large attendance which far exceeded that of last year's meeting.

Featuring the technical sessions were discussions of new developments and production increases. Production of iron powders, which is expected to quadruple during the next year, is considered a reflection of the wide acceptance of powder metallurgy in industry. Two new processes promise to aid in this expansion of industrial uses. "Precast" powders is one. The other is hot pressing, which, while only in the experimental stage, may be a method of fabricating precast alloy powders at presses far below those now required by cold pressing methods.

Stainless steel powder was described by John D. Dale, president of Charles Hardy, Inc., of New York. An interesting development shown in the product exhibit by this firm was a non-conducting permanent magnet, made by pressing metallic oxides in a relatively large amount of resin. Although it is claimed to have a magnetic strength comparable to Alnico, it has only about one-third the density. Immediate applications are seen in automobile speedometers.

Production of precision parts for the automotive industry was described by E. E. Ensign of the Ford Motor Company. Gears of extremely high accuracy were produced by a re-strike die which actually caused an extrusion process.

A special attraction during the meeting was the premiere showing of an Army Air Forces film on powder metallurgy. Product design practices as well as production techniques were explained.

At the product exhibit, small parts such as clock spacers, shown by the New Jersey Zinc Company, aroused much interest. These parts, formerly made of solid brass, are produced more economically and quickly from brass powder.

New president of the Society of the Plastics Industry—George H. Clark, vice president in charge of engineering of the Formica Insulation Company, Cincinnati; elected at the annual meeting of the Society of the Plastics Industry.

RECENTLY ELECTED SOCIETY OFFICERS

American Rocket Society, 29 West 39th St., New York 18, N. Y. President—Roy Healy; Vice President—Alfred Africano; Secretary—Leonard Axelrod; Treasurer—Cedric Giles.

American Society of Tool Engineers, 1666 Penobscot Bldg., Detroit 26, Mich. President—W. B. Peirce, Vice President, Flannery Bolt Co.; First Vice President—I. F. Holland, Pratt & Whitney Div., Niles-Bement-Pond Co.; Second Vice President—R. B. Douglas, industrial engineer; Third Vice President—W. A. Dawson, F. F. Barber Machinery Co.; Executive Secretary—V. H. Ericson, Johnson-de Vou, Inc.

Pressed Metal Institute, 122 East 42nd St., New York, N. Y. President—Tom J. Smith, Jr.; Vice President—Carter C. Higgins, Worcester Pressed Steel Co.; Secretary-Treasurer—J. J. Boehm, Boehm Pressed Steel Co.

Society of the Plastics Industry, 295 Madison Ave., New York, N. Y. President—George H. Clark, Formica Insulation Co.; Vice President—Gordon Brown, Bakelite Corp.; Secretary—Norman Anderson, General Molded Products, Inc.; Treasurer—Warren E. Hill, Prolon Plastics Div., Pro-phy-lactic Brush Co.; Board Chairman—Neil O. Broderson, Rochester Button Co.

MEETINGS

July 11–20
United Inventors and Scientists of America—World Inventors Exposition, Pan Pacific Auditorium, Los Angeles.

August 2-4
American Mathematical Society — First Symposium on Applied Mathematics, Brown University, Providence, R. I.

August 15-September 14
Canadian Mathematical Congress — Summer seminar, University of Toronto, Toronto, Canada.

August 21-22
Society of Automotive Engineers—West Coast Transportation and Maintenance Meeting, Biltmore Hotel, Los Angeles, Cal.

August 25-28
National Association of Power Engineers—Hotel Statler, Boston, Mass.

August 26-29
American Institute of Electrical Engineers—Pacific General Meeting, Hotel San Diego, San Diego, Cal.

August 28-September 13
British Engineers' Association—Engineering and Marine Exhibition, Olympia Bldg., London, England.

September 1-4
American Society of Mechanical Engineers—Fall Meeting, Hotel Utah, Salt Lake City, Utah.

September 2-5
American Mathematical Society — Summer meeting, Yale University, New Haven, Conn.

September 8-9
American Society of Mechanical Engineers — Industrial Instruments and Regulators Division, Chicago, Ill.

September 8-12
Instrument Society of America — Conference and Exhibit, Stevens Hotel, Exhibition Hall, Chicago, Ill.

September 15-19
Illuminating Engineering Society — National Technical Conference, New Orleans, La.

September 17-18
Society of Automotive Engineers—Tractor Meeting, Hotel Schroeder, Milwaukee, Wis.

September 17-26
National Machine Tool Builders' Association—Machine Tool Show, Dodge-Chicago Plant, Chicago, Ill.

Watt International Award To Timoshenko of Stanford

THE JAMES WATT INTERNATIONAL MEDAL, one of the highest international honors a mechanical engineer can receive, has been awarded to Dr. Stephen P. Timoshenko, professor emeritus of theoretical and applied mechanics at Stanford University. The medal was presented to Dr. Timoshenko recently in London, England, by the Council of the Institution of Mechanical Engineers of Great Britain. As a feature of the ceremonies, Dr. Timoshenko delivered a series of lectures to the Institution of Mechanical Engineers on "Stress Concentration and Fatigue Failures."

The James Watt International Medal, which is bestowed every two years, was founded by the Council of the Institution of Mechanical Engineers to commemorate the bicentenry of the birth of James Watt, who made many contributions to the development of the steam engine. Dr. Timoshenko and the late Henry Ford are the only Americans who have received the medal.

The Stanford scientist in 1935 was awarded the American Society of Mechanical Engineers Worcester Reed Warner medal, and in 1939 the Lamme Medal of the Society for the Promotion of Engineering Education. Dr. Timoshenko has been on the Stanford faculty since 1936.

DISCUSSIONS AND COMMENTS FROM READERS

Modified and Theoretical Formulas for Hinged Type Internal Blocks and Shoes

To the Editor:

This letter is written as comment on the article "Internal Friction Blocks and Shoes" by A. C. Rasmussen, consulting engineer, which appeared in the March issue of your publication. I regret to note that Mr. Rasmussen repeats and publicizes a basic error which is commonly made in the generally accepted method of analyzing these devices. The error consists mainly in neglecting to remember that forces have direction as well as magnitude, and consequently can only be handled by vectorial integration.

Where forces act normally on a flat surface, the vectorial integration of these forces reduces itself to a scalar integration, because the direction of all the forces is the same. However, when these forces, normal or otherwise, act on a curved surface, the plain scalar integration cannot be used any longer, and only vectorial integration will give the correct result.

In Mr. Rasmussen's analysis, departing as he does from the flat block analogy, we find that, in the case of the flat block, the resultant R, which is the scalar integration of the forces acting on the block, and the y_0 distance to what he calls the center or pressure, actually correspond to the truth for both quantities. When we go to the case of the curved shoe however, his resultant R of equation (17) is again only the scalar integration of the forces on the shoe. This value of R is basically wrong, and consequently we find that what he calls the center of pressure on the shoe no longer corresponds to the actual center of pressure (if we accept the definition of this expression as signifying the point where all the forces acting on the shoe on either side of this point engender self-balancing moments about the said point). This is caused by the fact that what he calls the center of pressure is of necessity derived from the moments equation about the anchor pin of the shoe, instead of about the center of pressure itself.

The consequences of this erroneous analysis appear mainly in the values given to the reactions of the anchor pin.

In the case of a fixed-anchor shoe, this faulty analysis is not very prejudicial to the proper functioning of the device. But the inherent weaknesses of this type of shoe are becoming more and more evident every day, and all designers in this field are endeavouring to get away from this fixed-anchor and substitute for it some kind of pivoted anchor, whether through a link or other means. It is here that it is of paramount importance that the correct pin reaction be found out, because it is this reaction which will have to play the all important function of centering the shoe on the drum and of insuring symmetrical distribution of pressure on the shoe lining. The analytical process involved in this type of shoe begins with the consideration of the shoe as of the fixed anchor type, so that one must know first how to properly analyze a shoe of the fixed anchor type in order to be able to proceed further.

It is no wonder that great disappointment has been met with this further development, since, on account of fallacious analysis, these devices have had the bad manners to perform quite differently from what was expected of them.

I have already written a paper on the subject, including a thorough mathematical analysis of these devices, which was presented at the A.S.M.E. Chicago convention, held June 16 to 19. I have often found, and this is a typical example, that it is not always entirely feasible to simplify a problem in order to solve it "with the least distraction by the mathematics."

—O. VON MEHREN
Project Engineer
The Thew Shovel Company

To the Editor:

In Mr. O. von Mehren's criticism of the article, "Internal Friction Blocks and Shoes," he points out that there is a difference between the hinged type shoe and the pivoted anchor or "floating" type of shoe. Although a general analysis can be developed to cover both types of shoes, the difference in action between the two types are sufficient to justify two different methods of approach. Consequently, the following reply refers only to the analysis for hinged type shoes.

Mr. von Mehren states that the value of R as given by the equation,

$$R = p_a br \left(\frac{\cos \theta_2 - \cos \theta_1}{\sin \theta_a} \right) \qquad (1)$$

is basically wrong. However this value of R is correct in representing only the value of the single force equal to the total normal pressure of the shoe against the drum. It does not include the force due to friction. Although the total frictional force, F, is equal to μR, the coordinates of F are not the same as those for R but are assumed to be the same for reasons explained later.

A hinged shoe of the type described in the article must be pictured always as a pivoted device in order that the analysis of its functioning may be kept clearly in mind. With this picture of the swinging shoe in mind, it is evident that the assumption can be made that until normal pressure on any small surface is directly proportional to the distance from the center of the hinge pin to the normal to that small surface. Further, with this assumption in mind, it is realized that the normal to a small surface that is farthest from the hinge pin is the normal to the tip of the shoe when θ_1 is 90 deg. or less. When θ_1 is greater than 90 deg., the normal to the tip of the shoe is nearer to the center of

the pivot pin than the normal at 90 deg. Consequently, when the shoe is 90 deg. or more in angular length, the maximum unit pressure is at 90 deg. from the hinge pin.

A pivoted shoe, when engaged, is held in engagement by two moments, one, the moment of the actuating force, T, and the other, the moment of the frictional drag of the drum on the shoe. With nearly all internal shoes, this frictional drag acts to press the shoe more firmly against the drum, causing in turn an increase in the frictional drag. With this self-energization, naturally, the result is that a lesser actuating force is necessary to obtain a desired frictional drag and hence, a desired coupling torque.

In actual friction devices of this type, there seems to be a loss of coupling effect with shoes more than 90 deg. in angular length and the loss becomes greater in proportion as the length increases toward 180 degrees.

A number of reasons may be given for this loss. With a longer shoe, the free particles, though minute, resulting from wear, act as a separating medium that reduces the friction of engagement toward the heel of the shoe. Outside dust and dirt, getting between the friction surfaces, aggravates this condition. Also with a heated shoe and drum, hot air and hot vapors from the lining tend to cause a separation of the friction surfaces. Should a hot lining tend to bleed, a reduction of the frictional drag will result. With a reduction of the friction of contact, a greater actuating force is necessary to produce a desired coupling torque. Likewise the load on the pivot pin becomes greater, although the greatest part of the load on the pivot pin comes from the coupling torque.

Formulas for the true analysis of a pivoted internal shoe are:

A—The total normal pressure of the shoe against the drum is,

$$R = p_a br \left(\frac{\cos \theta_2 - \cos \theta_1}{\sin \theta_a} \right) \quad (1)$$

The radius from the pivot center to the center of this total normal pressure is y_0, or

$$y_0 = a \sin \theta_0 \quad (2)$$

Therefore at the center of the drum, the angle between the pivot center and the radial line through the center of pressure is θ_0, obtained from

$$\sin \theta_0 = \frac{(2\theta_1 - \sin 2\theta_1) - (2\theta_2 - \sin 2\theta_2}{4 (\cos \theta_2 - \cos \theta_1)} \quad (3)$$

B—The frictional force on a small area is,

$$\Delta F = \mu p br \, \Delta \alpha \quad (4)$$

and as $p = \frac{p_a \sin \alpha}{\sin \theta_a}$ (5)

$$\Delta F = \frac{\mu p_a br}{\sin \theta_a} (\sin \alpha \, \Delta \alpha) \quad (6)$$

Integrating this equation between θ_1 and θ_2,

$$F = \mu p_a br \left(\frac{\cos \theta_2 - \cos \theta_1}{\sin \theta_a} \right) \quad (7)$$

From this equation it is seen that

$$F = \mu R \quad (8)$$

C—The moment arm of the frictional force on the small area is $r - a \cos \alpha$. Hence the moment of the force about the pivot pin center is

$$\Delta M_f = \frac{\mu p_a br}{\sin \theta_a} \Delta \alpha \, (r - a \cos \alpha) \sin \alpha \quad (9)$$

Hence

$$M_f = \frac{\mu p_a br}{\sin \theta_a} \int_{\theta_2}^{\theta_1} (r \sin \alpha - a \sin \alpha \cos \alpha) \, d\alpha \quad (10)$$

By integrating,

$$M_f = \mu \, p_a br \left[\frac{r (\cos \theta_2 - \cos \theta_1) - a/2 (\cos^2 \theta_2 - \cos^2 \theta_1)}{\sin \theta_a} \right] \quad (11)$$

$$\text{As } y_f = \frac{M_f}{\mu R} = (r - a \cos \theta_f) \quad (12)$$

Therefore

$$\cos \theta_f = \frac{\cos \theta_2 + \cos \theta_1}{2} \quad (13)$$

D—From the foregoing equations,

$$P_1 = R \sin \theta_0 + \mu R \cos \theta_f - T \sin \beta \quad (14)$$

$$P_2 = T \cos \beta + \mu R \sin \theta_f - R \cos \theta_0 \quad (15)$$

As a check, by taking moments about the center of rotation,

$$P_1 = \frac{Rr + T_z}{a} \quad (16)$$

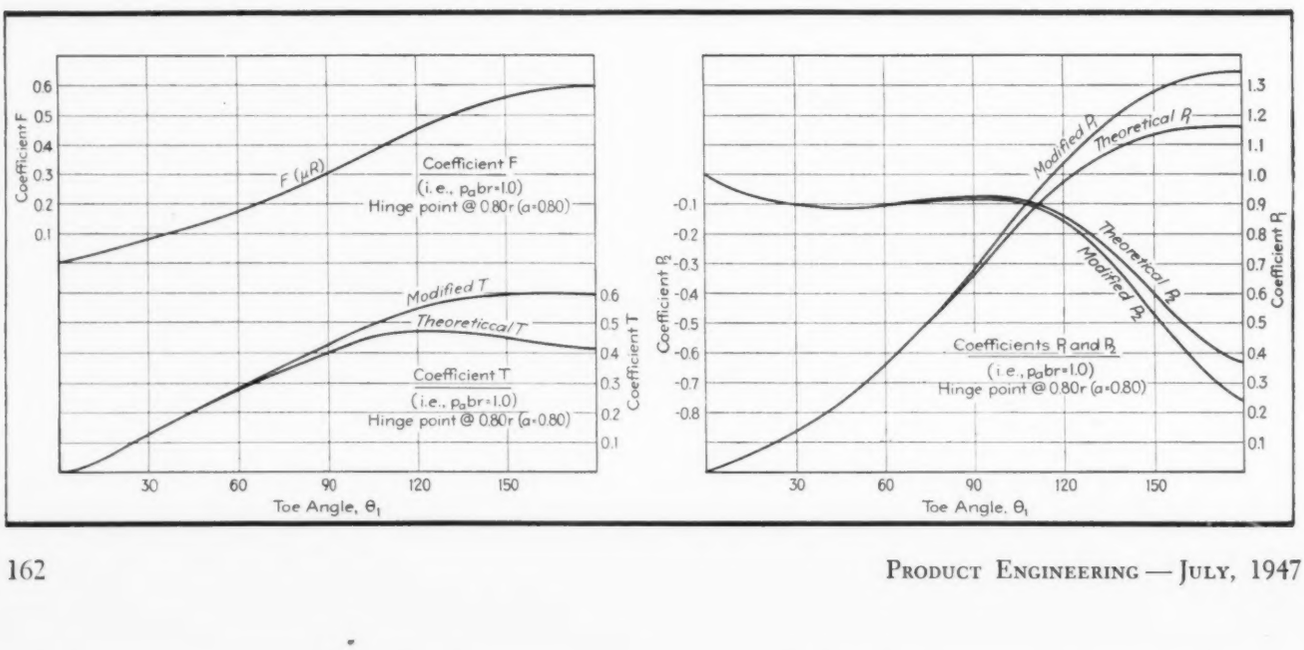
in which z is the radius of the force T, about the center of rotation.

Although θ_f is greater than θ_0 for all values of θ_1, θ_f has been taken as equal to θ_0 in order to modify the equations and allow for the reduction in friction toward the heel of the shoe on long shoes. This assumption does not result in a true value for P_2 but the discrepancy is not great.

In order to give a visual comparison of the results obtained by using the modified equations as against those obtained by using the true equations, graphs have been prepared. As the value of R is the same with both solutions therefore F will be the same. Consequently the coupling torque is the same. However with the modified solution, a greater actuating force, T, is required. Likewise there are differences between the values of P_1 and P_2 for the two solutions. It will be noted that the differences between the values of P_1 are greater than the differences between the values of P_2.

The charts are drawn using unit coefficient values. That is, the values of F, T, P_1 and P_2 are all figured with the value of $p_a br$ in the equations taken equal to unity. Also the hinge point radius, a, has been taken at $0.80r$.

—A. C. Rasmussen



Hydraulic O-Ring Packing For Sealing Cylinders and Pistons

ROBERT E. ALLEN
Plastic and Rubber Products, Inc.

Groove designs for non-moving and moving types of O-ring seals in hydraulic units and other applications. Typical arrangements are shown for sealing cylinders, pistons, end caps, valve covers and adjustable threaded parts. Sealing action of O-rings and properties of compounds available for making O-rings are described.

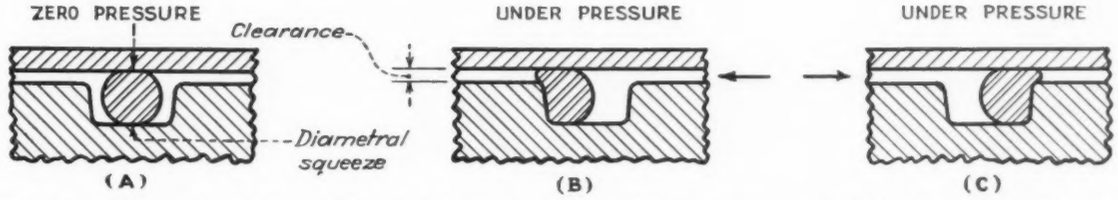


Fig. 1—Initial and working positions of O-ring seal in packing groove of a hydraulic piston. (A) At zero pressure. (B) When pressure is from the right side. (C) When pressure is from the left side.

IN MANY hydraulic systems, even a small amount of internal leakage may reduce efficiency and prevent proper operation of equipment. Also the loss of fluid from the system by external leakage may be of much importance, especially where reservoir capacities are made small to reduce weight. Fire and accident hazards caused by oil covered floors are other considerations. Hence, the selection of a reliable and simple packing for sealing the hydraulic fluid in a system is important.

Many designers have found that the use of O-ring seals, which were developed by the Air Forces during World War II, has many advantages that include: Simple construction in limited space without close fitting and space-requiring supporting structures; dependable sealing through wide ranges of pressure, temperature, and speed of motion; long life with a gradual increase in leakage until the seal wears away; easy assembly and replacement without special tools; and low cost.

The O-ring, Toroid or Donut seal is a circular ring of round cross-section, molded of synthetic rubber to close tolerances. In assembly, Fig. 1(*A*), the ring is fitted into a rectangular groove machined in the hydraulic mechanism.

At low fluid pressures the natural resilience of the compressed synthetic rubber furnishes the necessary sealing force. Under high fluid pressures the distortion of the cross-section augments the sealing force.

Under pressure the O-ring is forced away from the pressure side of the groove, Fig. 1(*B*). When the direction of the pressure changes, the ring rolls or slides over to the opposite side of the groove, Fig. 1(*C*). In motion, the sealing property of the O-ring comes from a rolling, sliding, wiping and squeezing action against the sealing surface like the action of a windshield wiper blade. The fluid being sealed provides lubrication.

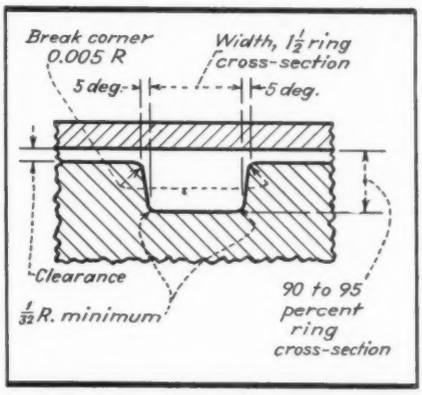


Fig. 2—Recommended design for groove.

Whenever possible, the groove should be machined on a rod rather than in a cylinder wall or a machined bore to facilitate machining, inspection and installation. The width of the groove at its bottom, Fig. 2(*A*), should be one and a half times the diameter of the ring cross-section. The groove should have 5 deg. sloping sides, so that in machining the tool can be run back without raking the side. To squeeze the ring, the depth of groove plus the clearance space between the surfaces to be sealed varies with the cross-section of the ring, ranging approximately from 10 percent for rings of 1/16 in. dia. cross-section to 5 percent for rings of 1/4 in. dia. cross-section. All groove surfaces should be free of nicks, burrs, and scratches.

Increasing the squeeze makes assembly more difficult, shortens the life of moving seals and increases friction. When a particular installation has more friction than desired, the depth of the groove should be increased until operating conditions indicate that a safe limit has been reached. Friction can also be de-

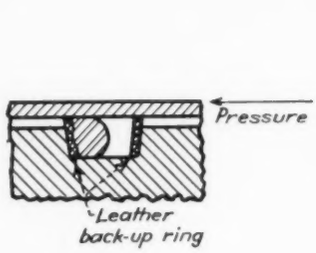


FIG. 3

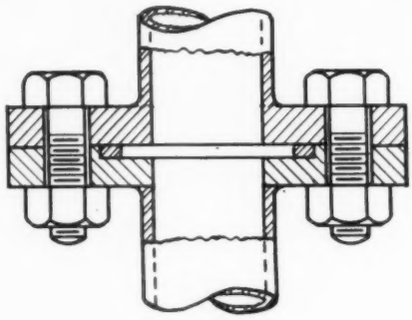


FIG. 4

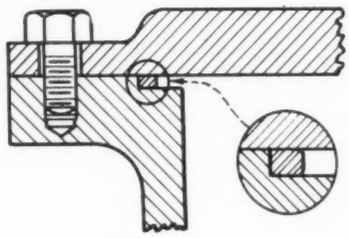


FIG. 5

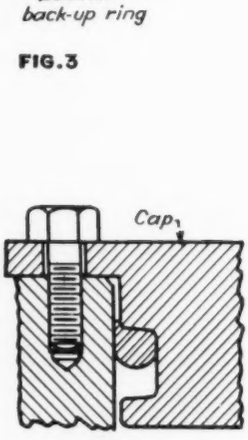


FIG. 6

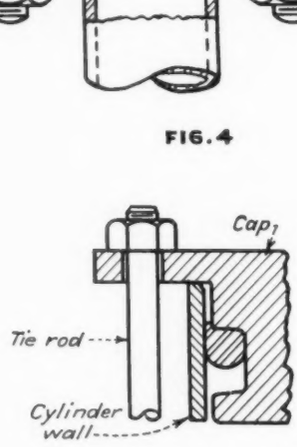


FIG. 7

Fig. 3—Groove fitted with leather back-up rings. Fig. 4—Flange pipe joint sealed with an O-ring. Fig. 5—Arrangement for sealing blind flanges and cylinder head covers. Fig. 6—Cap with O-ring used to seal the end of a bored hole. Fig. 7—Cylinder with end cap and tie rods.

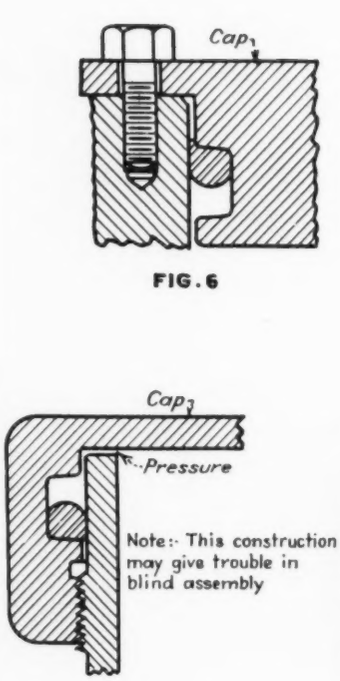


FIG. 8

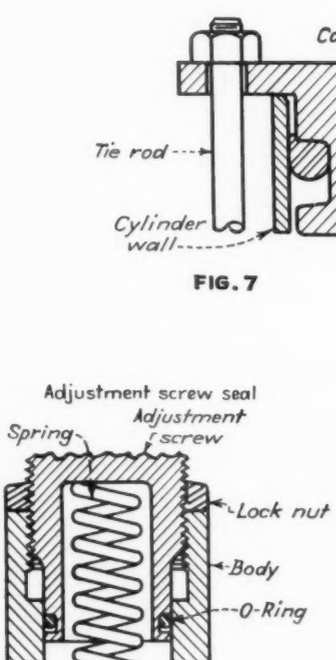


FIG. 9

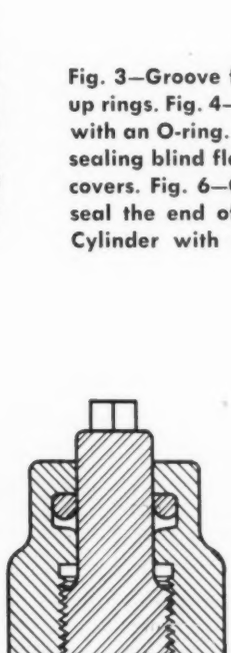


FIG. 10

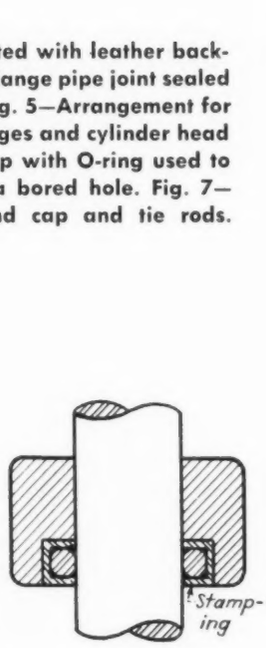


FIG. 11

Fig. 8—Cylinder with threaded end cap. Fig. 9—Method of sealing adjustable threaded parts. Fig. 10—Valve stem of the rising type sealed with an O-ring. Fig. 11—O-ring inclosed in a stamping seals a drill press spindle.

creased by using rings made of a softer compound, but at the risk of extruding the ring into the clearance space between the cylinder wall and piston. Decreasing the squeeze reduces the friction but may result in leakage at low pressures and low temperatures. The principal cause of failure of high pressure O-ring seals is the extrusion of the seal material through the clearance gap between the mating metal parts. The extent of extrusion depends upon: Clearance, fluid pressure, and durometer hardness of the O-ring compound.

Clearances should be held as close as production methods will permit. Design clearances should allow for the difference in the thermal expansion of the metals throughout the temperature range used to avoid binding. When using O-rings of 70 Shore "O" durometer hardness, without leather back-up rings, for satisfactory performance the maximum clearance between cylinder wall and piston at different pressures are:

Pressure, lb. per sq. in.	Maximum clearance, in. on a side
0 to 500	0.010 to 0.008
500 to 1,500	0.008 to 0.003
1,500 to 3,000	0.003 to 0.001

Many factors influence the running friction such as material and finish of the metal parts, oiliness of the hydraulic fluid, squeeze of the cross-section, and deformation of the ring under hydraulic pressure.

Running friction caused by the diametral squeeze of the ring increases with the hardness of compound, the percent of squeeze of cross-section and the length of rubbing surface.

Running friction caused by deformation of the cross-section under hydraulic pressure varies with the projected area of the seal exposed to pressure, and with hydraulic pressure.

In Table I are shown estimated

running friction loads for a ¾x1x⅛ in. ring used on a piston for sealing hydraulic oil of good lubricating value.

An anti-extrusion leather back-up ring on each side of an O-ring, as shown in Fig. 3, permits wide latitude in the clearances.

The use of leather back-up rings is recommended on all medium and large size pistons operating at pressures greater than 1,500 lb. per sq.in., because of the diametrical expansion of the cylinders. Unless extremely heavy tubing is used, the breathing or increase in diameter of the cylinder under high pressure will increase the design clearances and cause extrusion. For data on leather back-up ring sizes see "O-Ring Packings Adapated for 3,000-lb. Pressures", PRODUCT ENGINEERING, January 1945, p. 56.

In the non-moving types of seals, where friction is not a factor, the diametral squeeze on the ring may be much greater than that used for moving seals. An application of an O-ring to seal a flange joint is shown in Fig. 4. In this simple installation it is not necessary to machine a groove, only a recess in the face of one flange is machined. The diameter of the recess is that of the nominal O.D. of the O-ring. The flanges mate metal to metal, thus the strain on the bolts is less than that required to compress a gasket. With this type of joint, pressures as great as 20,000 lb. per sq. in. have been sealed satisfactorily, even under extreme vibration and repeated impulsing. The type of joint shown in Fig. 4 should not be used on vacuum lines.

A similar construction, Fig. 5, can be applied to blind flanges and cylinderhead covers. An end cap that deflects only slightly under load should be used to avoid extrusion. Fig. 6 shows how the O-ring can be used to seal the end of a bored hole. With many variations this construction can be used to seal cylinder ends, caps, plugs and valve covers.

A hydraulic cylinder made of a piece of tubing, fitted with end caps held in place with tie rods, and sealed with an O-ring is shown in Fig. 7. The tie rods are free to elongate and the cylinder barrel is free to expand and contract. The tie rods hold only the end pressure, therefore, the nuts need only be finger tight. The tie rods and nuts do not have a prestress as when flat compression gaskets are used.

A cylinder end cap of the threaded type with an O-ring is shown in Fig. 8. The blind assembly necessary with this construction may be difficult. But assembly can be facilitated if the O-ring is held in the groove with heavy cup grease and the outside corner of the cylinder barrel is chamfered to "shoe horn" instead of pinch the O-ring.

Sealing of adjustable threaded parts is shown in Fig. 9. This construction can be used on pressure relief valves, refrigeration expansion valves and similar devices.

Moving or dynamic seals fall into two classifications, those for oscillating or rotary motion and those for reciprocating motion. For high speed continuous rotary motion, O-ring seals should not be used because heat that cannot be dissipated is generated at the contact face. The heat burns and chars the contact surface, causing leakage and cracking of the compound. For slow speed rotary motion, O-rings can be used. See "Performance of O-Ring Packings in Rotary Seals for 3,000-lb. Pressure", PRODUCT ENGINEERING, March, 1945.

Table I—Total Seal Friction
Estimated for a ¾ by 1 by ⅛ in. ring

Pressure, lb. per sq. in.	Approximate Running Friction, lb. 70 Shore	80 Shore	90 Shore
0	4	7	9.5
250	8	11	13.5
500	11	14	16.5
1,000	17	20	22.5
2,000	24	27	29.5
3,000	29	32	34.5

The simplicity of the construction shown in Fig. 10 recommends the use of O-rings to the valve designer, wherever possible. Note the absence of stuffing boxes, multiple packing rings and gland adjusting devices. The design shown in Fig. 11, in which an O-ring is inclosed in a simple stamping, is for low pressure, slow speed rotative motion. This design has been used as an oil seal on spindles turning at 900 r.p.m. on drill presses.

There have been a number of applications of O-rings to face-type mechanical seals. The O-ring is ideal for sealing leakage along a shaft or a bore, since the ring will remain tight under slight lateral movements. A number of such designs are covered by mechanical seal patents or applications pending. So, search the prior art before designing a seal.

In reciprocating service, O-rings enable designers to make full use of space and cost savings. Fig. 12 shows a hydraulic cylinder in which four O-rings are used.

Under proper conditions of operation, the life of an O-ring depends upon: 1. Material and finish of rods and cylinder bores; 2. Material of piston heads; 3. Lubrication; 4. Protection from dirt and grit.

Smooth ground, chrome plated, steel piston rods should always be used if possible. Ground and honed steel

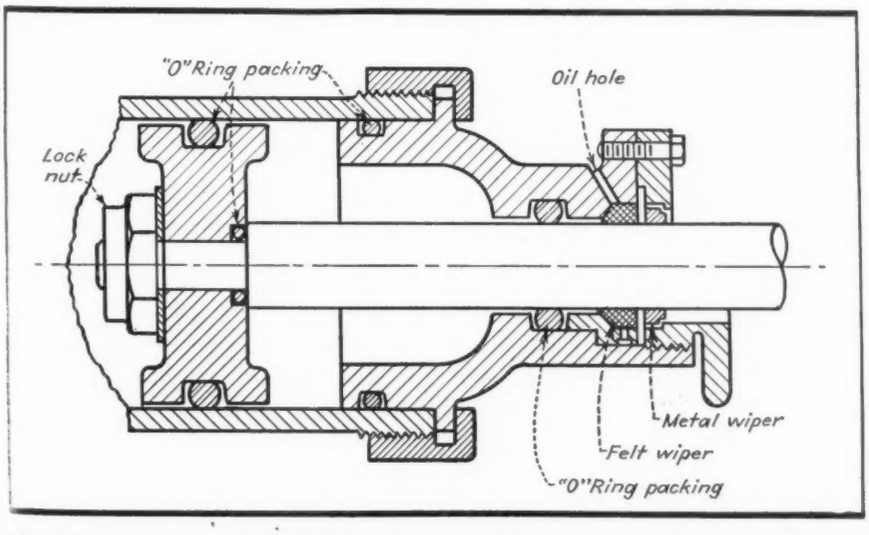


Fig. 12—Hydraulic cylinder head and reciprocating piston packed with O-rings.

or honed cast iron cylinder barrels should be used. To obtain maximum life, the surface finish should be as smooth as possible. A micro inch finish of from 5 to 15 rms., as measured with a profilometer, is recommended.

Some work has been done in sizing and burnishing commercial steel tubing. Drilled holes in cast iron are burnished by forcing selected size, hardened steel or Carboloy balls through the bore. The ball is always larger than the starting size of the hole. The burnished finish produced is excellent for O-ring applications. This technique makes honing unnecessary, it is cheap and produces a bore exactly to size and perfectly round. The best lubricant for the broaching operation is white lead paste. Carburizing steel or cast iron cylinder barrels will increase the life expectancy of O-rings. To preserve the surface finish of steel cylinder bores from scratches, hardened steel chrome plated pistons should be used.

Although they are satisfactory for some low pressure applications, aluminum alloy, bronze, brass, Monel and soft stainless steel cylinders are not generally recommended. Aluminum alloy, brass or bronze pistons may scratch steel cylinder bores.

Since O-rings wear rapidly when operated dry, when used in pneumatic systems both the ring and the groove should be lubricated with a high melting point grease. It is advisable to install a felt ring in a separate groove on each side of the O-ring. The felt should be saturated with a good grade of lubricating oil and oil should be regularly injected into the system.

If the O-ring operates in a hydraulic fluid of good lubricating value the fluid will provide sufficient lubrication. It is recommended that felt rings be installed on the outboard side of piston rod seals for lubrication. The felt should be thoroughly lubricated before assembly and a means provided for periodically adding lubricant through suitable oil holes. The felt should be protected from outside abrasives. Felt, oil and abrasives are a good lapping agent.

The hydraulic oil in the system should be absolutely clean and free of even micron size particles. All parts should be thoroughly cleaned of metal chips, sand, dust, or foreign matter before assembly. Piston rods should be protected with synthetic rubber boots to prevent sand, dirt and foreign matter from getting into the system. If it is not possible to use rubber boots, spring bronze scraper rings such as AN-6231 should be used.

Generally, the use of more than one O-ring per piston or rod is not recommended, because of the possibility of forming a pressure trap between the rings.

In assembly, O-rings should not be forced over threads, sharp corners, keyways or slots that might cut or nick the surface of the ring and thus cause leakage. All such sharp edges should be rounded, burred, or otherwise undercut to avoid damage. Chamfer all corners over which rings must slide when in place in the groove. Make the chamfer long enough and of such diameter that a "shoe horn" effect will be provided for easy assembly.

In operation, the O-ring should not be required to pass over corners, slots, ports or step diameters that will leave the ring unrestrained under pressure during any portion of the stroke. Such designs will cause cutting or pinching of the ring.

Experimental work has shown that, in some designs, the cross-section of the O-ring can be increased over the cross-section now regarded as standard to obtain better performance characteristics. This deviation is particularly true in the smaller cross-sections. The 1/16 and 3/32 in. nominal cross-section rings are critical when used as vibrating seals.

There are two series of O-rings: Those conforming in size to AN-6227 with cross-section 1/16 to ¼ in. nominal, for use as both moving and non-moving seals; and those conforming in size to AN-6230, all uniformly ⅛ in. nominal cross section in I.D. sizes 1-⅝ in. and larger, these should be used for non-moving seals only. For complete range of sizes and corresponding recommended groove dimensions see "O-Ring Packings Simplify Lightweight Hydraulic Equipment", PRODUCT ENGINEERING, September 1944, page 581.

Much can be said about compounds suitable for O-ring service. During the war all manufacturers were using the same compound, to a formula made available by the government. This special compound was developed for use with a special freeze resistant hydraulic oil that was suitable for use at temperatures of —65 deg. F. without breakage. This low temperature flexibility was obtained by sacrificing the physical properties on the high side of the temperature range.

Some engineers have used AN Grade O-rings at operating temperatures higher than 160 deg. F. with poor results, because the AN compound breaks down at higher temperatures. Fortunately, there are compounds available that are satisfactory for temperatures of as high as 250 deg. F. and that are flexible at —30 to —40 deg. F., as low as most industrial applications go.

For O-ring service, synthetic rubber compounds must possess: (1) Low compression and permanent set at the maximum operating temperature so as to maintain resilience and the diametral squeeze, (2) low swell in the hydraulic fluid used, (3) minimum loss of other physical properties under actual operating conditions, (4) no tackiness, sticking, corrosion or adhesion to the metals used and (5) maximum resistance to abrasion and wear.

Manufacturers of O-rings have developed special compounds with all of the five desirable qualities. The ordinary technical rubber specifications, such as ASTM and AMS do not go far enough in their requirements for O-ring usage. Designers should outline their problem to the manufacturer and accept his compound recommendations.

There has been much interest shown recently in the silicone type synthetic rubbers. In these polymers some of the carbon atoms have been replaced with silica. These rubbers are intriguing from a high temperature resistance standpoint but they do not, unfortunately at present, possess other physical properties suitable for O-ring usage.

The use of an O-ring in a groove is covered by U. S. Letters Patent No. 2,180,795, held by Mr. Niels A. Christensen. Arrangements have been made with certain O-ring manufacturers that when the O-ring is used in packing construction covered by this patent, licenses shall be available to all users through the manufacturers acting as agents.

FLEXIBLE CONTROL SYSTEMS

For Tandem Motor Operation

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Supervising Engineer, Cutler-Hammer, Inc.

Fundamental engineering considerations in applying a flexible adjustable-voltage control system to multimotor driven machines. Basic armature circuits, motor grouping, matching motor characteristics, range of speed under different conditions, and methods of speed synchronization, are some of the factors discussed.

ADJUSTABLE-voltage control is often used where a group of motors must operate together and stay in a desired speed relationship over a rather wide speed range. It is a simple and efficient method of controlling the speeds of a number of motors simultaneously by merely adjusting the voltage of a d.c. generator. The system is commonly applied for utilizing the inherent advantages of d.c. driving motors for adjustable speed applications in plants or installations that have an alternating current power supply but no direct current power lines.

Adjustable-voltage systems are well suited for driving machinery where the process requires a number of motors, each driving a separate section. The material being processed usually passes from one section to another in a continuous flow. By driving these motors from an adjustable-voltage generator, speed changes all along the process line can be made simultaneously. This type of drive usually requires some synchronizing means to keep the motor speeds automatically in the proper relation to each other. A group of motors operating in tandem in this manner is frequently referred to as a "range" drive.

Basic Armature Circuits

A simple adjustable-voltage armature or "loop" circuit diagram for a multi-motor drive is shown in Fig. 1. The shunt field and control circuits are omitted for the sake of simplicity. Motors No. 1 and No. 2 operate from the adjustable-voltage through a magnetic contactor, 1*M*. Each

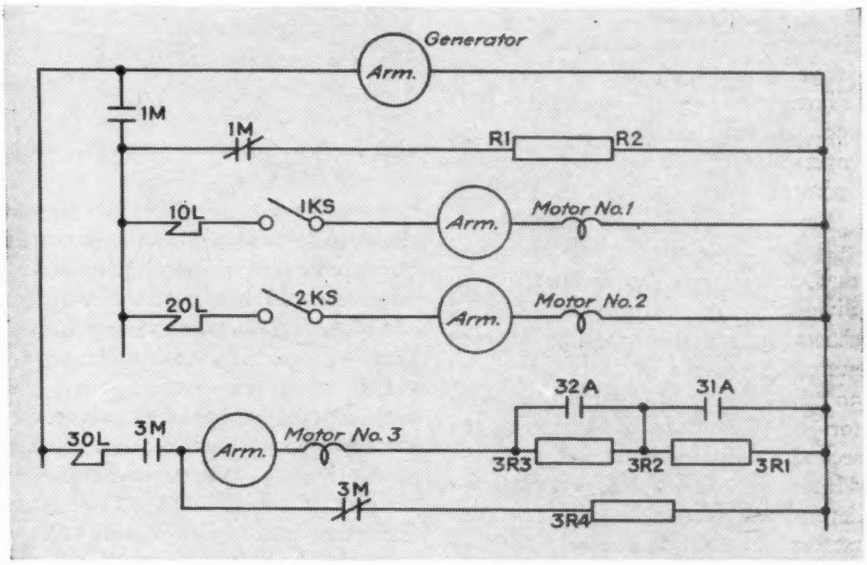


Fig. 1—Basic armature circuits of an adjustable voltage multimotor drive with one generator supplying three motors, no shunt fields and control circuits shown.

motor has its own overload relay *OL*, and isolating selector switch *KS*. A normally-closed contact on 1*M* provides a dynamic braking circuit through resistor R_1-R_2 for motors No. 1 and No. 2, as shown.

Since there is no starting resistor for motors No. 1 and No. 2, the generator voltage must be quite low when the contactor 1*M* closes to start the motors. To insure low voltage at starting, an interlock on the generator shunt-field rheostat is closed when the rheostat is in its lowest voltage position. This interlock prevents closing 1*M* except at low generator voltage. After 1*M* closes, the generator voltage can be increased to obtain the motor speed desired.

When the stop pushbutton drops out 1*M*, motors No. 1 and No. 2 are stopped by dynamic braking at a rate fixed by resistor R_1-R_3. Since both armatures are connected to a common dynamic resistor, their counter-voltages, and thus their speeds, must stay together during stopping. The speed of any number of motors can be held together during stopping by this method and it is usually used on range drives that process a continuous web of material. If these speeds were not held together during stopping, the material might pile up between sections or might tear from unequal stopping time of different motors.

In Fig. 1, motor No. 3 has a separate loop and dynamic braking contactor 3*M*. It has a conventional armature resistor with accelerating contactors 31*A* and 32*A*. This permits starting motor No. 3 at any

value of generator voltage and therefore it can be started and stopped while the other motors of the range are operating at the desired speed. Some processes require one or more motors arranged to operate in this manner.

Grouping of Motors

The sequence of the process and the process machinery usually determines the grouping of the motors in an adjustable-voltage range. Some machines require that all motors must be operated simultaneously during inching, running, and stopping. This is done with the simplest form of circuit having only one loop contactor and one dynamic braking circuit. Sometimes a manual switch is provided to disconnect certain motors from the range when they are not required. These are usually standard safety switches with warning plates on the covers stating that they are not to be operated when the unit is running.

Some manufacturing processes require grouping the various motors so that certain ones can be operated individually, or as groups. Fig. 2 shows the armature circuits of four motors connected in three groups to an adjustable-voltage generator. Motors No. 2 and No. 3 comprising the second group always operate together. Motors No. 1 and No. 4 operate independently from their own loop contactors. In addition, motor No. 4 has a manual-reversing switch for changing its direction of rotation.

In Fig. 2, all motors require low voltage starting. Each group has separate start and stop pushbuttons for individual operation. In addition to these, there is usually a master start and stop pushbutton to operate all groups simultaneously.

IR Drop Important Factor

When possible, motors for adjustable-voltage range drives are selected to follow each other in speed relationship as the speed is increased or decreased. This matching of motors requires that special attention be given to the resistance of the motor armature circuits, especially when a variety of frame sizes are used. Speed synchronization is simplified if the motors are selected so that all have about the same internal IR voltage drop when in operation, regardless of the

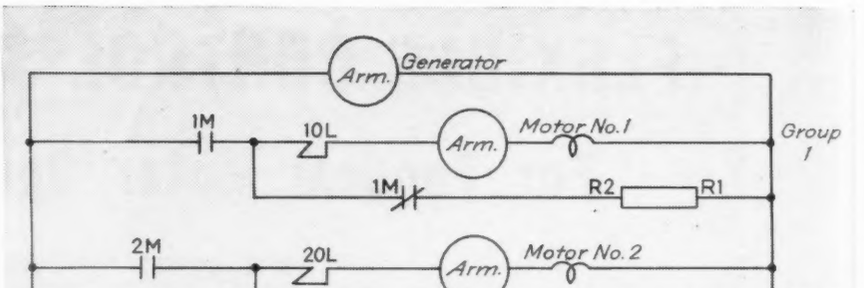


Fig. 2—Grouped motor circuits. Each group has individual start-stop control.

horsepower rating. The IR drop is the voltage produced by the armature current flowing through the internal resistance of the armature circuit.

When several motors are operated from a single adjustable voltage generator, their ability to stay in speed synchronism when the voltage is varied, depends upon the relationship of the IR drop in the various armature circuits. The IR voltage drop is an important factor in adjustable voltage wide speed range systems. When the IR voltage drop is relatively large, the motor performance is unstable at low generated voltage and thus at low speed.

The basic relationship is:

Voltage generated = motor counter e.m.f. + IR drop

For example, assume a motor must operate at 40 volts generated to get slow speed, and that it has an IR drop of 20 volts. This leaves only 20 volts for motor counter electromotive force.

If the load increases 25 percent, the IR drop is increased to 25 volts, which reduces the motor counter e.m.f. to 15 volts and the speed of the motor is correspondingly reduced one-fourth. In this example, the speed regulation of the motor at low voltage would be poor and would be especially noticeable when driving a reciprocating type machine having pulsations in the load.

This example illustrates the extremely poor speed regulation obtained at low voltage when the IR voltage drop of the motor is relatively large. In addition to the speed regulation considerations of the individual motors, the IR voltage drop affects the speed synchronization of a group of motors. To synchronize properly, the motors should be selected so that they have an armature circuit resistance equal to the same percent of E/I for each motor, where I is the load current of the individual motor and E is the common line voltage.

When a group of motors operating from a common generator have the same percent of E/I resistance in their armature circuits, the IR drop voltages are the same for all motors when carrying their rated loads. Since they have a common voltage supply, they all have the same counter e.m.f., and therefore the same relative speed and will operate together throughout the speed range.

IR Drop Compensation

There are occasions where it is not convenient to match all of the motors in a range drive to have equal armature circuit IR voltage drops. Successful operation requires additional elements to cause the motors to operate together properly. Some processing ranges or lines have certain motors that hold back rather than

drive to create tension in the web or strip. For this system, additional equipment also is needed to compensate for the different IR voltage drop conditions in the various motor armature circuits.

Compensation of the IR voltage drop is usually obtained by the use of small, low-voltage generators called "boosters." The booster generator armatures are connected in series with the motor armatures, and the booster voltage is adjusted in each to compensate for the IR voltage drop in each circuit. If the booster voltage equals the IR voltage drop for each motor, the counter e.m.f. of each motor will equal the main generator voltage and the motor speeds will all maintain the correct relation to each other when the main generator voltage is changed.

A typical adjustable-voltage loop circuit using booster for IR voltage drop compensation is shown in Fig. 3. The small booster generators are driven by an a.c. motor. The low voltage shunt fields of the booster generator are excited from the voltage drop across the motor commutating field windings. This gives booster compensation proportional to the commutating field voltage drop and therefore to the motor load. The amount of compensation is adjustable by a small rheostat in each booster field. The shunt fields of the booster generators can be connected to a constant-voltage supply and will operate successfully, especially if the motors are driving constant-torque loads. Although the small generators are called boosters, they may be connected so they actually buck the main generator voltage, such as when connected in a holdback motor circuit. With such a connection the motor actually regenerates back into the system and the IR voltage drop compensation must be reversed.

Most multimotor adjustable-voltage drives have some means of automatically synchronizing the speeds of the various motors with reference to each other. This speed correction is obtained by means of motor shunt field rheostats that provide vernier adjustment of the fields. This automatic synchronization is effective over the entire speed range of the drive and requires that the motors respond to the automatic devices at the lowest armature voltage.

To get motor shunt field response at a low generated voltage, the IR voltage drop must be relatively low. If the IR voltage drop is not low, it is possible that weakening the motor shunt field may result in a speed decrease rather than increase. For example, if a motor is operating at a low generated voltage and its shunt field is suddenly weakened, the immediate effect of the weakened field is to cause the motor to take more armature current in an attempt to accelerate. However, this increased armature current increases the IR voltage drop and this may be great enough to decrease the speed. The IR voltage drop must be extremely low if the motor is to respond properly to automatic control.

Speed Range by Adjustable Voltage

There are many factors that influence the degree of speed control obtainable by adjustable voltage. These include the size of motors selected, the nature of the load being driven, whether automatic speed synchronization is required at threading speed, and the possibility of using crawling speeds under ideal conditions.

In general, when the load is steady and when shunt field response of the motors is not required, dependable speed ranges of 8 to 1 or 10 to 1 by adjustable voltage are possible. When the load may fluctuate, the dependable range is about 6 to 1 or less.

When motor shunt field response is required, the speed range obtainable is determined by the IR voltage drop, as discussed previously. The lowest operating speed can be determined approximately by the rule that the IR voltage drop in the motor must not exceed 10 percent of the lowest applied voltage. This is particularly true of sectional paper making machines where accurate speed synchronization is required and certain sections may be driven by motors of twice the power required by the running load in order to keep the IR drop low.

On machines that require an extremely wide speed range, adjustable speed motors can be used to extend the speed range obtainable by adjustable voltage. Overall speed ranges of 20 to 1 or even 30 to 1, under favorable conditions can be obtained.

Acceleration and Deceleration

Range drives, in general, require a slow rate of acceleration and deceleration. This is necessary to enable the various motors to stay in the proper speed relation to each other while the speed is being changed. In manual systems the rate of change is under the control of the operator. Motor-operated rheostats are well suited for regulated acceleration, since it is possible to adjust the speed of the rheostat pilot motor to obtain the desired rate of acceleration.

On simple range drives, the motor-operated rheostat is usually controlled from fast and slow hold-down type

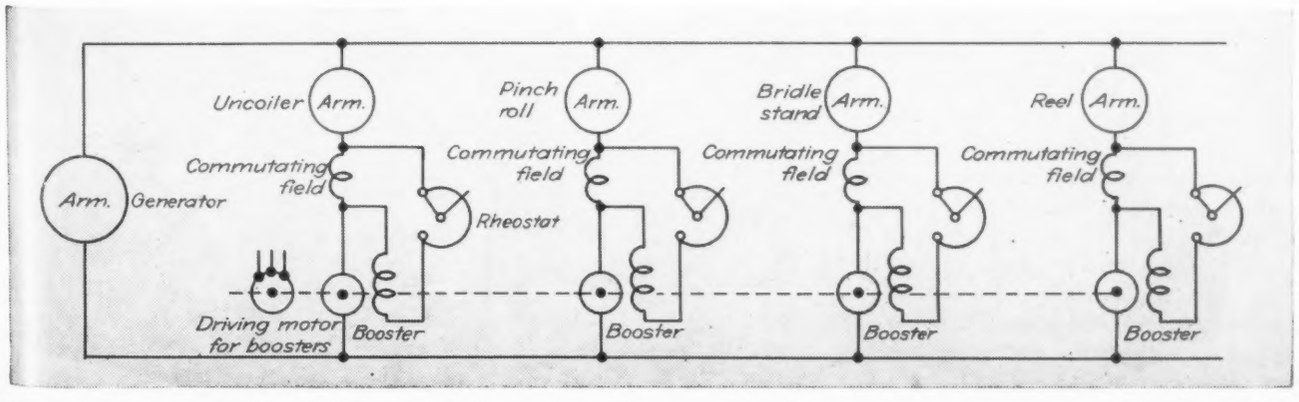


Fig. 3—Low voltage boosters for IR voltage drop compensation are excited from voltage drops across commutating fields.

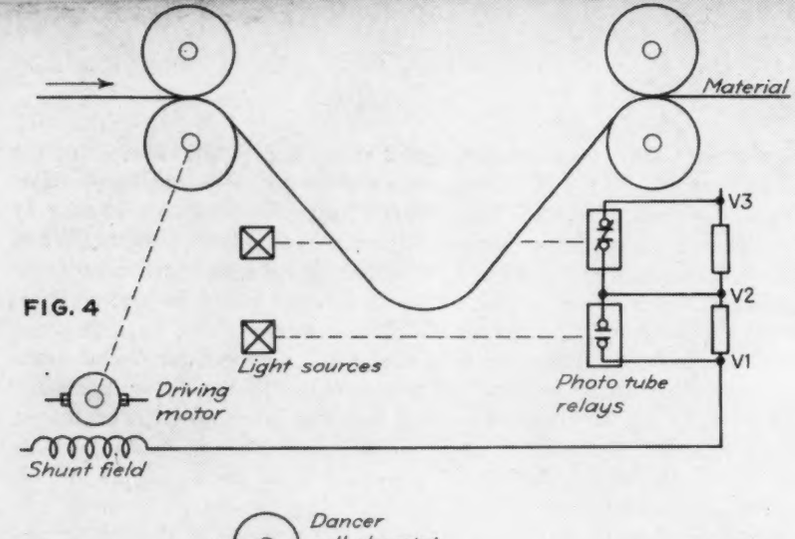


FIG. 4

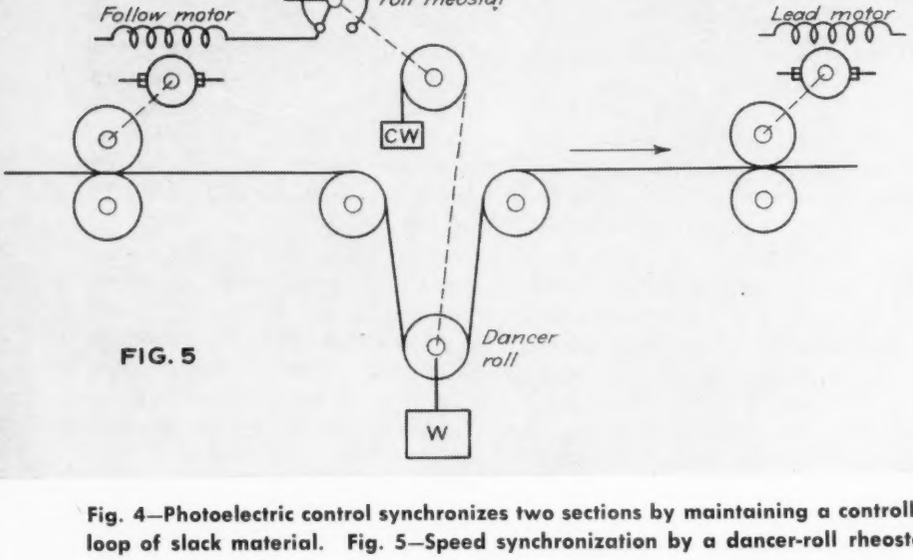


FIG. 5

Fig. 4—Photoelectric control synchronizes two sections by maintaining a controlled loop of slack material. Fig. 5—Speed synchronization by a dancer-roll rheostat.

pushbuttons. By manipulating these buttons, the speed is adjusted to the desired value. Occasional applications require a motor-operated rheostat for controlled acceleration with a preset speed adjustment. If the preset running speed does not need to be changed frequently, it can be set by means of an adjustable high speed limit on the motor-operated rheostat. The circuits can be arranged so the machine will automatically accelerate to the preset speed when the start or fast pushbutton is operated.

Sometimes the preset speed must be easily adjusted from a remote point. A simple method of doing this is to use a preset field rheostat and an automatic field accelerating device. A more elaborate method uses a polarized relay followup scheme in conjunction with a motor-operated rheostat. This requires a potentiometer-connected manual speed presetting rheostat for the remote control station and a similar potentiometer follow-up circuit on the motor-operated generator field rheostat.

Dynamic braking is commonly used for stopping. In some applications the range is slowed down by regenerative braking and then stopped by dynamic braking. Some schemes use only regenerative braking for stopping. When the stop pushbutton is pressed, the motor-operated rheostat returns to the low voltage position at high speed, during which time the regenerative braking torque is appreciable. The final stop is then made by the control circuit applying a suicide connection to the generator.

Sometimes, a quick regenerative braking stop is made by simply using the time-constant of the generator field to govern the braking torque. When a stop is made, the field is disconnected from the line and discharged through a resistor circuit to obtain the proper rate of generator flux decay for the stopping effort required.

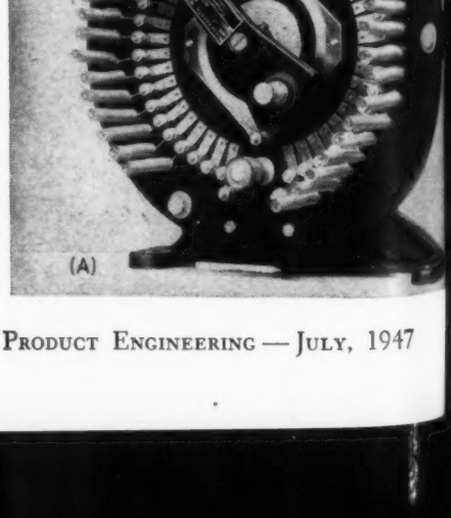
Speed Synchronization

When a continuous strip or web of material pases from one machine section to another in a process, the speeds of the various motors driving the machine sections must be accurately synchronized. Machines of this type are usually called range drives and are commonly used in the textile industry. The accuracy of speed control required depends upon the process and upon the strength of the material being processed.

Frequently the speeds of the various section motors are adjusted manually by means of vernier field rheostats. This is satisfactory on low speed applications and on processes where the material is strong enough so that there is little danger of breakage. Typical of these are some strip steel processes, the manufacture of roofing paper, and the processing of cardboard. It is important to match the section motors as regards IR voltage drop when manual control is used. Each section motor has an individual manual vernier rheostat. An ammeter in each motor circuit is a simple and relatively inexpensive way to help the operator maintain proper speed adjustments.

Sometimes the process requires a loop or loose festoon of the material between sections. This may be either a metallic or fabric strip. A convenient way of synchronizing two such sections is to use photocells and light beams to control the length of the loop or festoon, as shown in Fig. 4. In this sketch, two light beams are used and as the photo relays respond to these beams they control resistance in the motor shunt field circuit to cause the loop to stay within the zone of the two light beams at all times.

Dancer-roll rheostats have been long used in range drives and are still used where the material permits the use of a dancer roll. In the tex-



tile plants, this mechanism is called a "compensating gate". It consists of two fixed rolls and a movable roll known as the dancer roll as shown in Fig. 5. The compensating gate structure guides the dancer roll in vertical slots and provides a chain mechanism between the dancer roll and the dancer-roll rheostat. Any change in the length of the festoon of material rotates the dancer-roll rheostat and adjusts the field circuit of the following motor. This keeps the motor speed properly synchronized with the section ahead. In addition to the speed control obtained, the compensating gate also controls the tension in the web of material. The dancer roll is weighted and the dancer-roll rheostat is counterweighted. The difference in these weights represents the amount of pull or tension produced in the web.

Dancer-roll rheostats used in adjustable-voltage range drives usually have enough resistance to increase the motor speed 20 to 50 percent. The rheostats should be designed for easy operation and for extremely long life. A bracket mounted dancer roll rheostat having all contact parts and resistors easily accessible is shown in Fig. 6(*A*). The driving sprocket is on a shaft extending to the rear of the assembly, and is usually connected to the dancer roll by means of a length of roller chain.

Sectional paper making machines require close speed control so that each section can be maintained at a desired speed relationship, or draw, to the master section. After the speed relationships have been established, they must be maintained very closely.

The master section that sets the paper making machine speed, can be a constant speed motor-generator set or a constant d.c. voltage supply. The speed of any other section of the machine is then compared with the master speed and any differences are automatically detected by the control that instantly acts to correct the discrepancies. One system in wide use, uses a mechanical differential at each section. Opposite shafts of the differential are driven by the controlled section and the master section. Any difference in speed cause the differential cage to rotate and that in turn rotates a field rheostat connected in the section motor shunt field. This corrects the section motor to the proper synchronizing speed.

Paper machine sections are also controlled by voltage comparison methods. The section speed is measured by the output voltage of a small d.c. generator that is driven by the section. This d.c. generator voltage is compared with a reference d.c. voltage from a master generator or other constant-voltage source. Any differences in these voltages are detected and amplified electronically and used to change the section motor shunt field strength to bring the voltages into agreement.

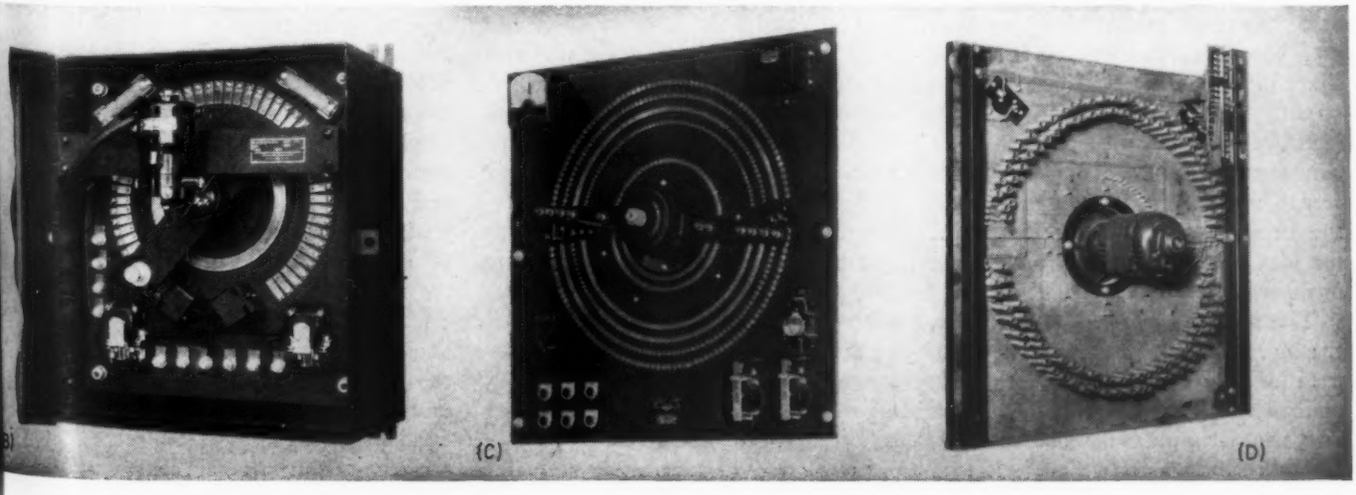
Motor-Operated Rheostats

The performance of an adjustable voltage system is often largely dependent upon the motor-operated rheostat. The prime purpose of this rheostat is to commutate the generator shunt field resistor and thereby regulate the generator voltage. If this is the only requirement, a single circuit device such as in Fig. 6(*B*) can be used. It can be a larger rheostat having more steps but its essential parts are a potentiometer-connected field rheostat and a pilot motor. The pilot motor is coupled to the rheostat lever and the circuits are arranged so it will drive the lever in either direction in response to remotely located "increase" and "decrease" push-buttons.

More elaborate control schemes require motor-operated rheostats having additional commutating circuits. These circuits can operate in generator and motor fields as required by the application, or can connect into the control circuits to operate control devices at the proper speed level. The construction shown in Fig. 6(*C*) is often used in many applications because of its flexibility. As many as nine concentric circles of contact buttons and sectors can be used and commutated simultaneously by sliding contacts carried on the rotating arm. A rheostat such as this can usually handle all of the circuits necessary for a multimotor range drive.

The motor and built-in gear reducer that drives the rheostat lever is mounted on the rear of panel as shown in Fig. 7(*D*). The motor responds to the control devices to produce the operating speeds required. For instance, proper acceleration and deceleration may require full travel of the rheostat in 60 seconds. However, if a quick stop is initiated requiring regenerative braking of the

Fig. 6—Typical rheostats used for regulating generator field circuits in multimotor control. (A) Heavy duty dancer-roll type. (B) Single circuit motor operated. (C) Multicircuit motor operated. (D) Rear of 120-step gearmotor operated rheostat.



adjustable-voltage system, the rheostat may need to travel from the high speed position to the off-position in 3 or 4 seconds. A slip clutch between the motor drive and the rheostat lever absorbs the shock when the lever arm hits the off position stop at high speed. By mounting field circuit resistor units directly on the studs that hold the contact buttons and segments, the usual wiring between contacts and resistor is not necessary and the resistors and other back-of-panel parts are more easily accessible for inspection and servicing.

The foregoing basic principles and limitations that designers should keep in mind when applying adjustable voltage to multimotor driven machines point out in a general way the range of possibilities offered by this extremely flexible system. As a result of long use in diversified engineering fields, new ideas and refinements are constantly being added to make adjustable-voltage control for multimotor drives even more useful to designers. Its widely accepted application on an increasing scale on many types of machines and in almost every industry, is substantial proof that it fulfills a performance need more economically than other methods.

Experimental Stress Analysis

F. R. STEINBACHER
University of Illinois

MUCH OF THE THEORY upon which modern structures are designed is based on mathematical analysis. To make the mathematics simple or easy to handle, broad assumptions are often made that are true only in certain limited ranges. The most obvious use of theory beyond the range for which it was intended that comes to mind is the indiscriminate use of the My/I formula to find the bending stress regardless of its range. The standard method for solving indeterminate structures also deserves some attention. Bending beyond the elastic range has received much attention lately, but little study has been given to the broader field of indeterminate structures in which members are stressed beyond the elastic limit.

In an indeterminate structure the loads carried by its members are in proportion to their stiffness. The stiffness of a member is directly proportional to the applied load up to a certain point. Beyond this point the stiffness is no longer linear. Thus other members not stressed as high may be obliged to carry an increased share of the load. As the load on a structure is further increased, its distribution among the resisting members of the structure changes. As different members are stressed beyond the linear range, the distribution of the load continues to vary among the members.

The more complex the structure is, the more difficult it becomes to make a rational analysis in which the calculated stresses agree with the actual. A typical example is the airplane with its thin shell construction, stiffened by light longitudinal and transverse members. The airplane structure is highly redundant especially in the regions of doors, windows, engine nacelles and landing gear cutouts. When such a structure is subjected to a complicated set of loads from cargo, passengers, pressure differentials, and inertia, it is formidable if not impossible to develop a theory that will predict an accurate stress pattern.

Since accurate stress calculations are exceedingly difficult to make for many designs, manufacturers recognize the importance of proof testing and willingly set aside much time and money to prove their products thoroughly. Hundreds of small units are tested during the design stage before the complete product is loaded. Since production is often well underway by the time the final tests are completed, the results of all the tests are watched closely. Small changes can often be made to improve the design before final assembly.

The results of a continuous test program can be used to revise methods of analysis with the view of predicting more accurately the behavior of similar products in the future. This procedure has led to a much clearer understanding of the behavior of reinforced thin shell construction. The problem of shear lag in wide thin sheets, the analysis of light rings supported by thin shells, the calculation of deflection of frames in non-circular pressure cabins are three outstanding design problems that have profited by recent testing research.

Stress analysis testing plays an important role in the design of many products. But before a product or machine can be tested it must be designed and built. The stress analyst with the aid of all the theory, experience, and information available must determine to the best of his ability the loads on the various members throughout the machine. He estimates the size of the members and often suggests to the designer some revision in the type of structure or a change in the location of certain members so that the loads can be carried more efficiently. The correlation between the analysis and the test results is a measure of the stress analyst's ability. A conscientious and competent analyst is quite proud when his calculations compare favorably with the test. As his experience increases, he becomes more and more capable of predicting the behavior of complex structures.

The test engineer studies the behavior of general types of construction. He also determines how much load new designs and new materials will hold, and how they behave under different types of load. Some of the manufacturer's best engineers are employed in test work. They draw curves, devise formulas, collect data, and study theory so that the stress analyst can better predict the action of new designs or the behavior of the new material. The problems encountered have interested many scientists and their work in correlating theory with tests has led to solutions that enable some of the highly redundant structures to be placed in the more conventional class.

NEW BOOKS

Mathematical Aids for Engineers

RAYMOND W. DULL. 346 *pages, 5¼x8¼ in. Green clothboard covers. Published by the McGraw-Hill Book Company, Inc., 330 W. 42nd Street, New York 18, N. Y. Price* $4.50.

It is the purpose of this book to supply engineers, and those preparing to become engineers, with the basic mathematical aids essential to the solution of problems associated with the many new engineering developments.

The fundamental principles that underlie mathematical operations are thoroughly analyzed, and their application is illustrated by many examples. The contents of the book can be appraised by a few of the chapter headings such as: Uniform scales, logarithms as an implement, circles as tools, vectors, imaginary and complex numbers, hyperbolic functions, kinetics of rotation, and kinetics of plane motion. There are also five chapters dealing with differential equations.

More than 250 line cuts are used in the book to illustrate the discussion.

The Chemistry Of Commercial Plastics

REGINALD L. WAKEMAN. 836 *pages, 6 x 9¼ in., clothboard covers. Published by Reinhold Publishing Corporation, 330 W. 42nd St., New York* 18, *N. Y. Price* $10.

This book is not a "pure chemistry" treatise, but rather correlates theory with actual industrial practice. A comprehensive work, this volume contains much information of use to the design engineer. In addition to the chemical discussions, properties and applications of molding powders, finished and semi-finished shapes, and laminates are covered. Production methods, including molding, laminating, casting, die-pressing and blanking, blowing, coating, extrusion and low pressure molding are thoroughly treated.

Precision Hole Location

J. ROBERT MOORE. 448 *pages, 7 x 10 in., clothboard covers. Published by The Moore Special Tool Company, Bridgeport, Conn. Price* $5. *Reduced prices to the metal-working industry.*

For interchangeability in toolmaking and production, the designer finds much information in this treatise about the location and production of holes with accuracy and economy. Seven chapters of text review the evolution of hole location practices to the point where the tool maker can use engineering methods and apply the principle of interchangeability in the making of jigs and fixtures. The text is illustrated with over 300 photographs and diagrams.

Also included in the book are circular tables of coordinate factors and angles by W. J. and J. D. Woodworth. These tables give the rectangular coordinates for locating from 3 to 100 holes, inclusive, on the circumference of a circle.

The Theory of Functions Of Real Variables

LAWRENCE M. GRAVES. 300 *pages, 6 x 9 in., red clothboard covers. Published by the McGraw-Hill Book Company, Inc., 330 W. 42nd Street, New York* 18, *N. Y. Price* $4.

An exposition of the more fundamental and generally useful parts of the theory of functions of real variables, this book also contains many useful results not generally found in the standard treatises on the subject, such as some theorems on implicit functions, differential equations, and Lebesque and Stieltjes integrals.

A general appraisal of the contents of the book can be gained from some of the chapter headings such as: The real number system, point sets, functions and their limits, fundamental theorems on differentiation, the Riemann integral, uniform convergence, functions defined implicitly, and ordinary differential equations.

Elements of Mechanism

SCHWAMB, MERRILL *and* JAMES. *Sixth edition, revised and rewritten by Venton L. Doughtie. 428 pages, 6 x 9 in., clothboard covers. Published by John Wiley & Sons, Inc., 440 Fourth Avenue, New York, N. Y. Price* $4.

The Two aims have been kept in mind in revising this well-established textbook; namely, the retention of the presentation of the fundamentals and the use of current terminology, notation, standards, illustrations and examples. Chapters III, IV and V treat vectors, velocity analysis and acceleration analysis. Many new problems have been included, all of which may be worked on 8½ x 11 in. paper.

The sequence of chapters has been changed for better continuity. Two chapters of the previous edition have been combined. Chapter XIII, Miscellaneous Mechanisms, contains some of the matter previously presented in another chapter.

Fundamentals of Industrial Electronics Circuits

WALTHER RICHTER, *Consulting Electrical Engineer, Allis-Chalmers Mfg., Co. First edition, 569 pages, 6¼ x 9¼ in., clothboard covers. Published by McGraw-Hill Book Co., 330 W. 42nd St., New York* 18, *N. Y. Price* $4.50.

Fundamental principles are stressed rather than specific applications. Electronic circuits are reduced to a combination of more familiar circuit elements so that the average designer or electrical engineer can readily understand them. Mathematical treatment is held to a minimum.

The twenty-seven chapters include discussions of circuit elements, rectifier circuits, basic tubes, amplifiers, tuned circuits, fundamental control circuits and optical systems.

Elementary Heat Power

HARRY L. SOLBERG, ORVILLE C. CROMER *and* ALBERT R. SPALDING. 480 *pages, 5¼ x 8¼ in., clothboard covers. Published by John Wiley & Sons, Inc., 440 Fourth Ave., New York, N. Y. Price* $4.75.

Written for students who have had a course in college chemistry and who have completed at least one semester of college physics including heat and mechanics, the objectives of this book are:

1. To develop an understanding of the functions, principles of construction, and actual performance of heat-power machinery so that the student will appreciate the importance of and may intelligently study and apply the principles of engineering thermodynamics.

2. To provide an adequate and balanced terminal course for those students who do not study engineering thermodynamics.

3. To provide a background for the testing of heat-power equipment in the laboratory.

Emphasis throughout the book is placed upon the energy balance and the law of conservation of energy applied to prime movers. Problems on material and energy balances are included. Use is made of the characteristics equation of the perfect gas with which the student is familiar from physics and chemistry. Steam tables are used in the determination of enthalpy and specific volumes.

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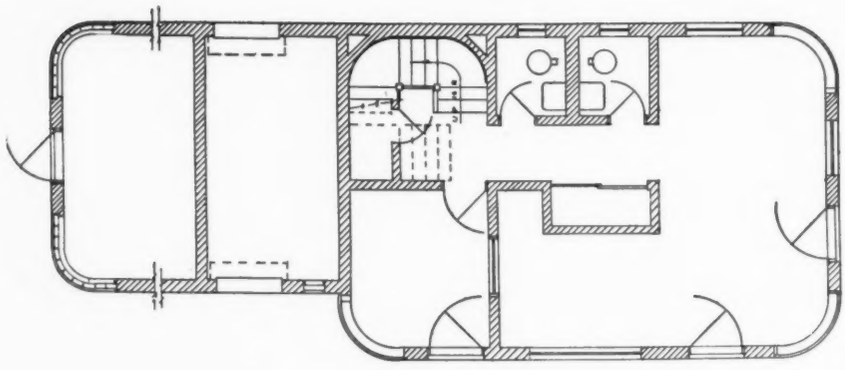

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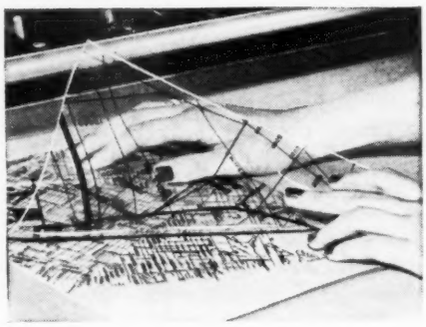

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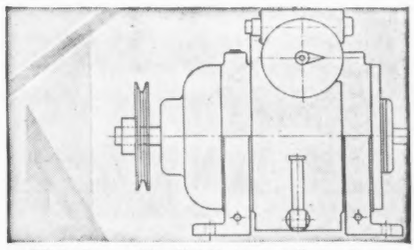

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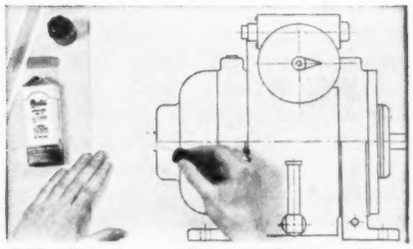

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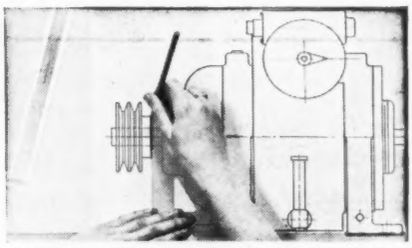

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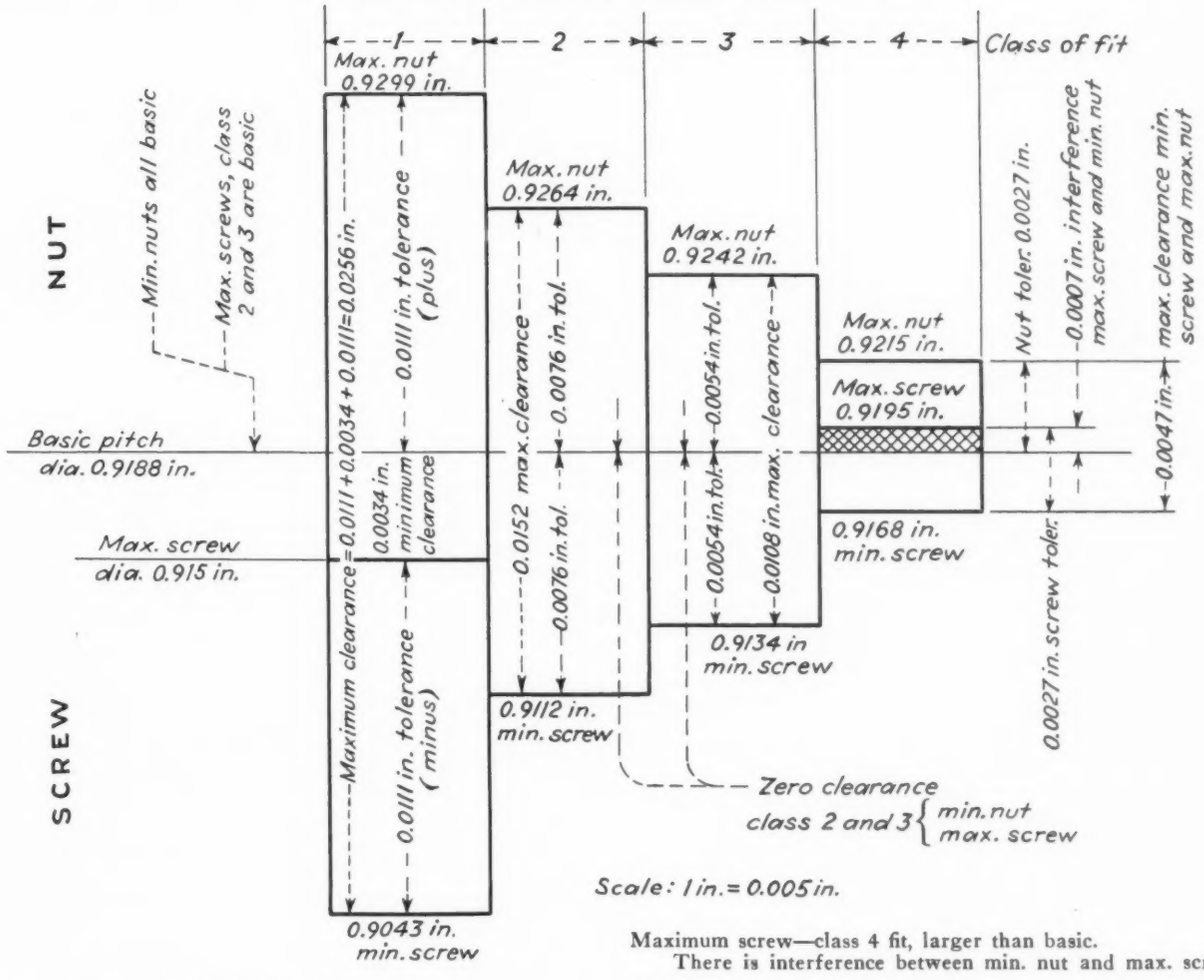
Scale Diagram for Relations of Thread Dimensions

JOHN L. SKEEHAN
Development Engineer, Treadwell Construction Company

The major diameter of the thread of a screw or nut is its largest, finished diameter. The minor diameter of the thread is its smallest diameter. The pitch diameter is the average between the major and the minor, and is equal to the major diameter minus the depth of one thread. The pitch diameter is the most important dimension as it governs the class of fit. The allowances and tolerances for the major and minor diameters are from two to three times those for the pitch diameter. The basic diameter is the theoretically correct diameter of a screw or nut and is used for designating the major, minor and pitch diameters. The maximum and minimum diameters give the permissible deviation from the basic measurements specified on working drawings.

This diagram illustrates the relations, allowances and tolerances of the pitch diameters of class 1, 2, 3 and 4 fits for a 1 in. diameter, 8 thread (1"-8NC) bolt and nut combination. It is proportionally representative of a large range of sizes of American National Course threads of various pitches and classes.

Scale Diagram of Pitch Diameter, Allowances and Tolerances
1"–8NC Screw Threads, Classes 1, 2, 3, and 4



Minimum nuts—basic size for all classes of fit.

Maximum screw—class 1 fit, less than basic.
There is clearance between min. nut and max. screw.

Maximum screws—class 2 and 3 fits, are basic size.
Metal to metal fit between min. nut and max. screw.

Maximum screw—class 4 fit, larger than basic.
There is interference between min. nut and max. screw.

Class 1 fit, with minimum tolerance is closer fit than class 4 fit with maximum tolerance.

Maximum nuts are rarely found because commercial tolerances for U.S. Standard hand taps, cut threads for 1 in., 8-threads are: 0.9228 in. max. pitch diameter and 0.9198 in. min. pitch diameter.

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FACTORY—BOONTON, New Jersey

Nomogram for Polar Moment of Inertia by Suspension

HERBERT F. BARIFFI
Professional Engineer

THE POLAR MOMENT OF INERTIA for an irregular mass, such as an armature or fan blade assembly, may be determined by suspending it from a beam or other fixed support. A slender, unstranded wire should be rigidly attached to the support and to the axis of rotation of the mass. This gives the effect of a torsional pendulum.

Using a steel wire of diameter d in., measure l in., the length of wire between the support and the hanging mass. Start the mass oscillating through a small rotational amplitude taking care not to exceed the elastic limit of the wire. A swinging, gravity-pendulum effect must be avoided. Measure the time, t sec., required to make 10 complete oscillations.

On the nomogram draw a line connecting the points for the values of t and d used. From the intersection of this line and the vertical turning line, draw a line to the value of l used. Where this line crosses the vertical scale of I_g read the polar moment of inertia for the mass.

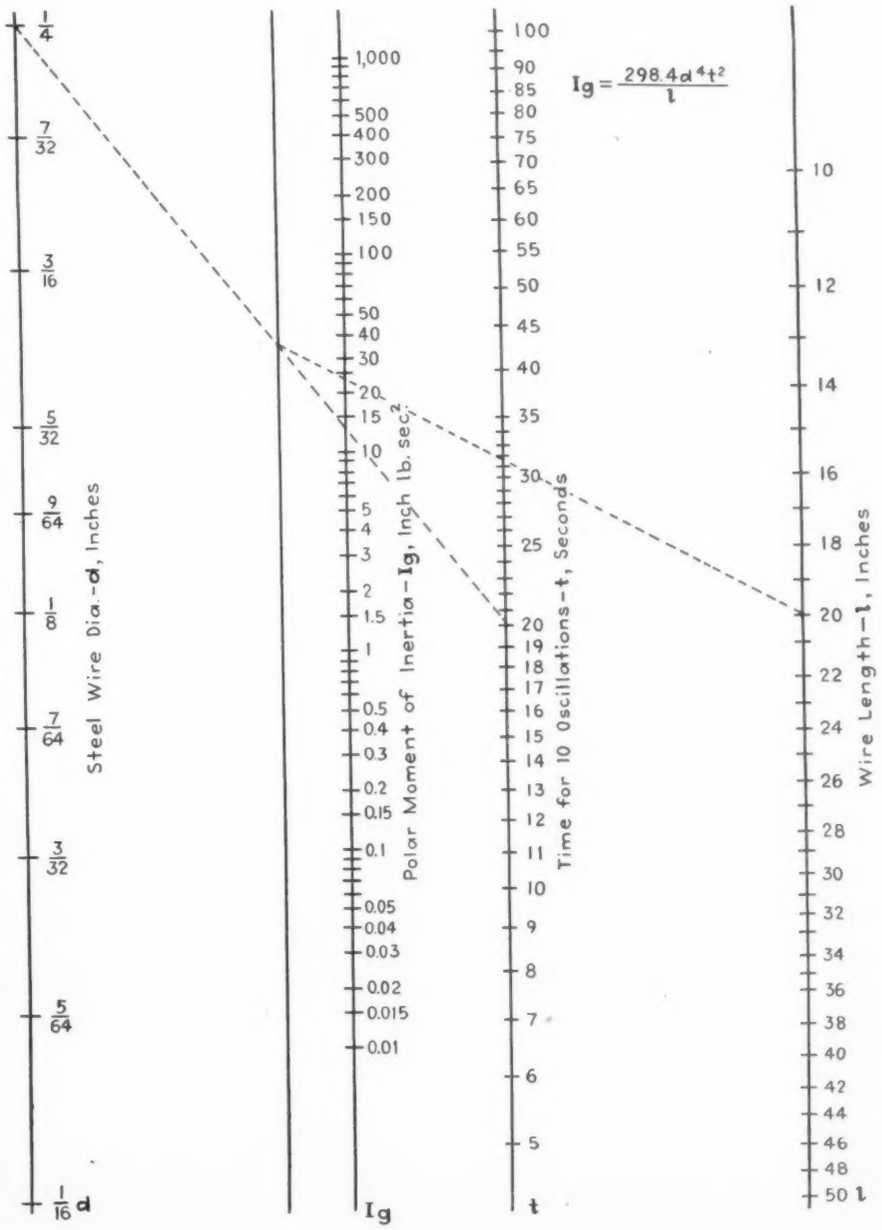
PROBLEM: Given steel wire with a diameter of $\frac{1}{4}$ in., length of 20 in. between support and mass of unknown I_g, find the polar moment of inertia if the time for 10 oscillations is 20 seconds.

SOLUTION: Draw line between $\frac{1}{4}$ in. on the d scale and 20 sec. on the t scale. From the point of intersection on the turning line draw a line to 20 in. on the l scale. Read 23.5 in. lb. sec. sq. for the polar moment of inertia.

This nomogram can also be used to determine the critical torsional frequency of a shaft-and-mass combination when the total polar moment of inertia is known. Connect shaft length l in. with polar moment I_g and produce to meet the turning line. From this point draw a line to the shaft diameter, d in. Where this line crosses the vertical time scale read t sec. This value divided into 10 will give the critical frequency in cycles per second.

This method of determining inertia by torsional oscillation neglects the mass inertia of the suspension wire. It is evident that even with a large wire the effect is negligible because of the usual large I_g value of the object and because, while the wire is slightly twisted at the lower end, it is stationary at the upper end.

The nomogram has been constructed using steel wire, for which the shear modulus of elasticity was taken as 12,000,000, and t as the time required to complete 10 oscillations.



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NEW MATERIALS AND PARTS

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General Testing Meter

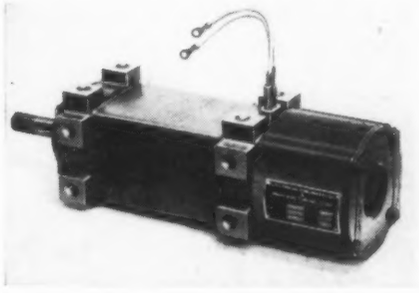
General Electric Co., Schenectady 5, N. Y.

The AF-1 a.c. load visualizer is a lightweight, low cost analyzer for use on single phase and balanced polyphase systems. It has an iron-vane moving element and separate field windings for voltage and current. The current winding is electrically insulated from the imposed voltage by an internal double current transformer. The current and voltage terminals are energized simultaneously from the line or through instrument transformers to extend the range. Operation of three selector knobs does not interrupt the external current circuit. The instrument is housed in a greaseproof black Textolite case and the potential terminals, instrument mechanism and control switches are mounted on a Textolite top plate. It is magnetically damped, fully shielded and equipped with an external zero set. This instrument serves as a 0-2.5 and 0-5 amp. ammeter and a 0-150, 0-300 and 0-600 volt voltmeter. It can also be used with the calculator furnished to determine watts, vars, volt-amperes, power factor for load surveys, induction motor tests, reactive power studies and power factor checks on power and lighting circuits.

High Speed Electric Motor

Electrical Eng. and Mfg. Corp., 4006 W. Jefferson Blvd., Los Angeles, Calif.

Built for direct connection applications, Eemco universal type electric motors, are totally inclosed and fan cooled. Aluminum castings, high temperature varnishes and glass-fiber insulation are used in the motor construction. Weighing 22 lb., this motor is 14¾ in. long, including shaft extensions, with a 5 in. diameter. It is a continuous duty type operating on 115 volts, a.c. or d.c., with 25 or 60 cycle current. The Eemco motor has a 5,700 rev. per min. running speed.



Conveyor Belt

United States Rubber Co., Rockefeller Center, N. Y.

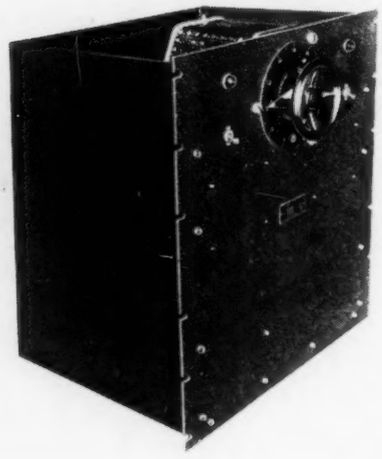
This rubber fabric belt is claimed to be 250 to 400 percent stronger than previous types. The increased strength is gained by a textile construction of nylon and Ustex yarn that permits more plies and increases the working

tension of each ply two and one half times previous allowable values. A long belt with a minimum of stretch and troughing qualities is possible because of low stretch lengthwise and increased flexibility crosswise.

Power Supply

Superior Electric Co., Church St., Bristol, Conn.

Built for continuous duty, these Seco d.c. power supplys give an adjustable output voltage with small regulation. They use a full wave bridge rectifier with four type 866/866A tubes. Time

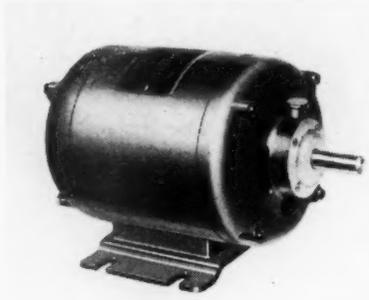


fuses are used to prevent damage from overload. One model, see cut, has an output range of 0–3,000 volts d.c. with a maximum current of 0.5 amp. It operates from a 115 volt, single phase, 50 to 60 cycle source. This power supply can be mounted on a standard 19 in. relay rack.

Synchronous Motors

Cyclohm Motor Corp., 5-17 46th Rd., Long Island City, N. Y.

Known as the Cyclohm 29 Size, these motors are designed for recording tape pulling and facsimile use. The 29 Size motors are normally supplied as balanced capacitor start and run types. They are two-phase motors with uniformly rotating magnetic fields. The frame ring, and front and back caps are die-cast. Stator and rotor laminates are punched from silicon steel, and heavy former insulated wire is used throughout. Synchronizing torque is obtained in the rotor through internal slots punched in the rotor laminations. Cyclohm motors can be supplied with either sealed and shielded ball bearings that are permanently lubricated, or with large self-lubricating bronze sleeve

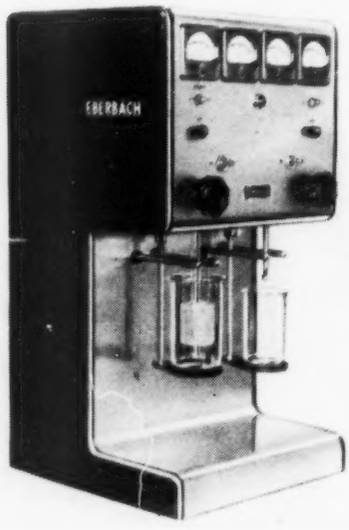


bearings. The latter are lubricated by a wool wick that cannot become glazed because it is not in contact with the shaft. Running at 1,800 rev. per min., these motors are available in ratings of 1/100, 1/75 and 1/50 hp. at 115 volts and 60 cycles.

Electro-Analysis Apparatus

Eberbach and Son Co., Ann Arbor, Mich.

The 32-560 Eberbach Electro-Analysis apparatus can be used by industrial or research testing laboratories for determinations of nickel, copper, lead, antimony, cadmium, zinc, chromium and other metals. Front surfaces are heavy gauge stainless steel and sides are black wrinkle coated cast aluminum. Meters and mechanism are contained within the unit for protection from fumes. A selenium dry type full wave rectifier delivers a current of 5 amp. at 8 volts d.c. at both spindles for simultaneous determina-

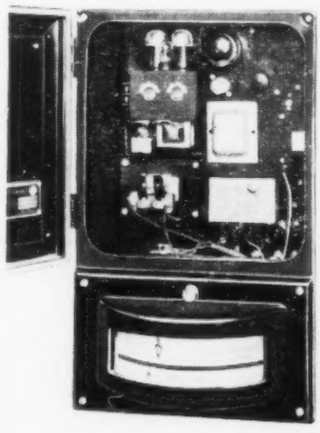


tions. A current of 10 amp. d.c. is delivered for single determinations. Direction of current flow, which is automatically indicated by the voltmeters, is controlled by polarity reversing switches. Stirrers rotate at a constant speed of 550 rev. per min. and are adjustable for height as are the breaker supports. All electrodes including rotating types can be used. This unit operates on a 110 volt or 220 volt, single phase, 60 cycle a.c. power source with 400 watts maximum input. Overall height is 29½ in., with a table area of 14½ by 16½ inches.

Temperature Control

Claud S. Gordon Co., 3000 S. Wallace St., Chicago 16, Ill.

A precision temperature-variation control instrument, known as the Xactline Capacitrol, is built to control temperature within ±½ deg. F., and power "on-off" cycles as short as 3 seconds. There are no cams, gears or motors and the unit will give instantaneous, electronic control. The instrument has a wide control range and a direct reading indicating scale. The Xactline Capacitrol is available in two, flush mounting types: One with two-position micro-switch control for a current of



10 amp. at 115 volts or 5 amp. at 230 volts, the other with "on-off" mecury-switch control for a current of 50 amp. at 115, or 25 amp. at 230 volts.

Aircraft Trim Tab Control

Airborne Accessories Corp., 25 Montgomery St., Hillside 5, N. J.

Using a principle of varying mechanical advantage, which increases with load, this motorized actuator for tab control can be used for various sized wings. Connected directly to the tab with a self-aligning link, it needs no additional screw jack, flexible cable or other transmission. Known as "Trim-Trol", this unit is rated at 1/100 hp., which is claimed to be much lower than control